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TECHNICAL REPORT

70-18-GP

AN ANALYTICAL AND EXPERIMENTAL STUDY TO DEVELOP A NONMECHANICAL SYSTEM TO INDUCE RESONANCE IN A ROD DRILL FOR FROZEN SOIL

by

Seth R. Goldstein

Foster-Miller Associates, Inc.

Waltham, Massachusetts

Contract No. DAAG17-68-C-0028

June 1969

General Equipment & Packaging Laboratory

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General Equipment & Packaging Laboratory
U. S. ARMY NATICK LABORATORIES
Natick, Massachusetts

FOREWORD

This analytical and experimental study to develop a non-mechanical system to induce resonance in a rod drill for frozen soil was prepared by Foster-Miller Associates under an exploratory development Project 1J662708D503-05-003 for the U. S. Army Natick Laboratories, Natick, Massachusetts. Previous work was conducted for the U. S. Army Natick Laboratories under Contract DA19-129-AMC-47(X) during the period from January 1963 to February 1965.

Mr. Jack M. Siegel of the General Equipment & Packaging Laboratory at the Natick Laboratories was the Project Officer for this study. Mr. John S. Howland was the Program Manager and Dr. Seth R. Goldstein was the principal investigator for Foster-Miller Associates. Dr. Herbert H. Richardson, Associate Professor at Massachusetts Institute of Technology, acted as a consultant to Foster-Miller Associates during this program. Appreciation is expressed to Dr. Roger M. Stinchfield, Physical Scientist, Engineering Sciences Division, General Equipment & Packaging Laboratory, U. S. Army Natick Laboratories, for his review and analysis of this report which resulted in many valuable comments and recommendations, and to Messrs. Albert A. Carletti, Chief, Shelters Branch, Shelters and Organizational Equipment Division, and Constantin J. Monego, Textile Technologist, Shelters and Organizational Equipment Division, U. S. Army Natick Laboratories, for their encouragement and support of this work.

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ABSTRACT

The objective was to investigate nonmechanical methods for inducing longitudinal resonant vibrations in a rod which could serve as a drill for frozen soil, and to select the most promising method for further development.

An analytical model of the mechanics of the vibratory drill was developed to determine the exciter output requirements. A large number of exciter concepts were conceived and evaluated.

The major conclusion is that a hydraulic exciter system is the best overall concept for meeting the exciter output requirements, and its feasibility has been established both analytically and experimentally.

Piezoelectric and magnetostrictive transducers are also feasible but are less desirable than hydraulic systems when weight, durability, safety, cost, and other factors are considered. Unstable bearings, electromagnetic devices, and various other concepts were examined and rejected for reasons of low efficiency and excessive size.

The best application of the hydraulic exciter concept utilizes a rotary valve to control flow to a piston-type actuator that is rigidly attached to the drill rod. A laboratory model of this type of exciter was designed, built, and tested to demonstrate the hydraulic excitation principle. The model system, which used a 6-foot long steel drill rod, exhibited a resonant amplitude peaking of four at a frequency of 1425 cycles per second.

AN ANALYTICAL AND EXPERIMENTAL STUDY TO DEVELOP A NONMECHANICAL SYSTEM TO INDUCE RESONANCE IN A ROD DRILL FOR FROZEN SOIL

1. INTRODUCTION

This report summarizes a study to investigate nonmechanical methods of producing a resonant vibration in a drill rod. A laboratory model drill rod exciter was developed and successfully tested.

Objectives

The ultimate objective is to develop a rugged, portable, self-contained vibratory drilling system that will enable tent stakes to be emplaced in frozen soil. The preliminary objectives were to investigate methods of inducing resonance in a drill rod by other than mechanical means, and to produce a laboratory model that demonstrates the most promising concept.

Background

A rod that vibrates in its free-free resonant mode can be used to drill holes in frozen soil. Since portability considerations limit the rod length to 72 inches, the resonant vibration frequency is approximately 1425 cycles per second (cps). In a study performed for the U. S. Army Natick Laboratories by the Arthur D. Little Company of Cambridge, Massachusetts(1), a 10-horsepower (hp), mechanically driven exciter was built. This device drove a 1-inch-diameter pipe into frozen soil to a depth of 8 inches in 15 seconds. In subsequent tests, the device experienced repeated mechanical failures.

The key element of the drilling system is the device which excites the drill rod into resonance, and which transmits to it the energy that is required to displace the drilled material. Previous exciters that have been designed have utilized purely mechanical means to transmit the required high-power, high-frequency vibrations. These devices (e.g., the Arthur D. Little exciter) have had insufficient durability to be incorporated into an operational system. Consequently, to avoid problems associated with mechanical breakages, a program was initiated to develop a nonmechanical drill rod exciter. The present report describes the first phase of this program.

Scope of Work

The following major tasks were performed:

a. development of an analytical model of the vibratory drill mechanics, and from it a determination of the required drill rod exciter force, stroke, and power;

b. generation of concepts for exciting the drill rod;

c. evaluation of the various exciter concepts, and a determination of their feasibility;

d. evaluation of the auxiliary equipment requirements of those concepts which are feasible, and a selection of the most promising drilling system; and

e. design, fabrication, and testing of a laboratory model of the most promising drill rod exciter concept.

2. SUMMARY

This program consisted of studies to generate and evaluate concepts for utilizing nonmechanical means to maintain a longitudinal resonant vibration in a drill rod.

In the initial phases of the work, an analytical model of the mechanics of the vibratory drill was developed and used to determine the output requirements of the drill rod exciter. It was assumed that the rod is to be excited close to the vibration node that exists near the center of its length.

For a typical drilling task, the exciter output requirements were predicted to be:

average power	-	5 hp
exciter stroke amplitude	-	± 0.003 in.
exciter force amplitude	-	± 5000 lb.
excitation frequency (aluminum or steel drill rod)	-	1390 to 1425 cps

In successive phases of the work, various exciter concepts were generated. The feasibility of these concepts was determined by their ability to produce the above typical outputs.

The major conclusion is that a hydraulic exciter system is the best overall concept for meeting the exciter output requirements, and its feasibility has been established analytically and experimentally.

Such a system customarily utilizes some form of modulator to control the flow of fluid to and from a piston-type of actuator that is rigidly attached to the drill rod. Due to the high working pressures that can be used in hydraulic systems (e.g., 2500 psi), the required high force levels can be achieved with a system that is lightweight and compact. A high-frequency excitation can be obtained by controlling the fluid with a rapidly spinning rotary valve that has a multiplicity of flow ports communicating with the actuator. The operator can tune the excitation frequency to coincide with the drill rod resonance by varying the valve rotational speed. The feasibility of this concept was experimentally demonstrated with a laboratory model that excites a 70-inch long steel drill rod into resonance at 1415 cps. At the resonant condition, the peak-to-peak amplitude of the drill tip vibration increased by a factor of four to 0.004 in. The band width of the resonance was approximately 40 cps. It is felt that a more pronounced resonance could be achieved if the system were redesigned to eliminate fluid inertia and resistive effects in the lines connecting the valve and the actuator.

A different type of hydraulic exciter system that was analytically determined to be feasible would utilize a specially designed hydraulic gear pump to generate pressure pulses that are fed to a piston-type of actuator. This modified pump combines the functions of the power supply and the fluid modulator element. However, due to the substantial amount of development work that would be required to perfect this technique, the concept was not considered further.

Piezoelectric and magnetostrictive transducers were analytically determined to be feasible to excite the drill rod. However, considerations of their required auxiliary equipment, weight, development and production cost, safety, and other operational factors indicated that they were inferior to a hydraulic exciter.

Pneumatic actuators, unstable fluid bearings, hydrodynamic oscillators, and fluidic devices were investigated. These devices were found to have undesirably low efficiencies, and they were not considered further. Devices utilizing fluids with special properties that can be electrically modulated were judged to be impractical at the present time. Electromagnetic and electrostatic devices were also judged to be impractical due to the excessive size that results from their low effective force per unit area.

Based upon the results of the present study, it is recommended that further development of the resonant drilling system be concentrated upon hydraulic exciter systems.

3. EXCITER SPECIFICATIONS

a. System Specifications

It is desired to develop a portable, rugged drilling system that can be handled by one man. This requirement limits the system weight to approximately 75 pounds and the drill rod length to approximately six feet. The system is to be capable of drilling a one-inch diameter hole at a rate of one foot per second to a depth of 30 inches.

b. Exciter Force, Stroke and Power Requirements

In Appendix A, page 29, the exciter frequency, force, stroke, and power requirements are established as a function of drilling rate, hole size, permafrost compressive strength, and system geometry. Representative exciter requirements which were used for concept evaluation are

exciter force	-	$\pm$ 5000 lb.
exciter stroke	-	$\pm$ 0.003 in.
average power	-	5 hp
excitation frequency	-	1425 cps

4. EXCITER CONCEPT GENERATION

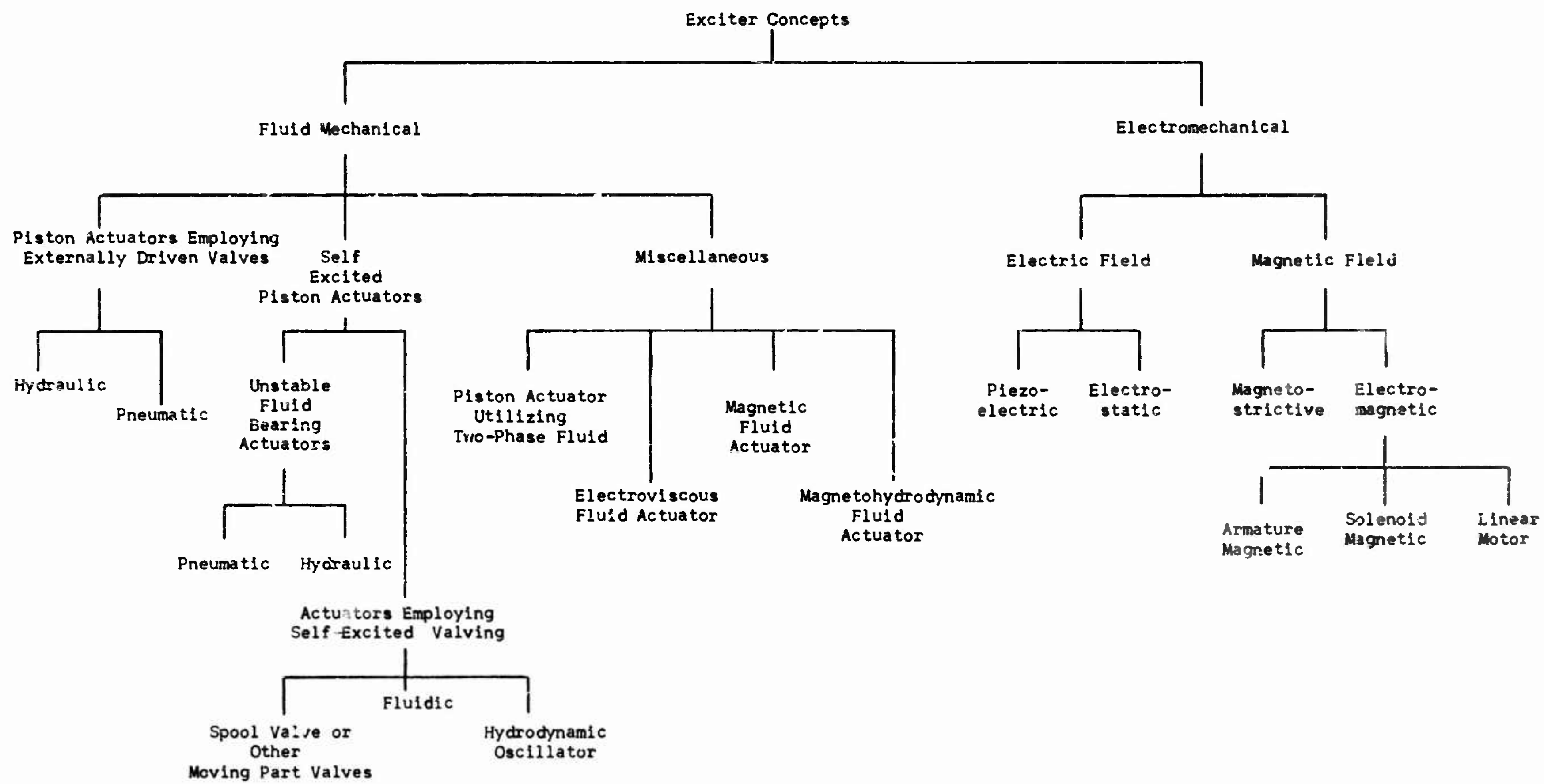
The first phase of this program involved the generation of non-mechanical exciter concepts. Novel and conventional techniques were suggested and grouped in various categories. Particular emphasis was given to those concepts for rod excitation which had the potential of small size, light weight, and high durability.

The results of this effort are presented in Table I, where the various exciters fall into the general categories of fluid-mechanical and electro-mechanical devices. Within the fluid-mechanical category a natural division exists between externally-driven fluid valving and self-excited fluid valving. Both types of systems employ positive displacement actuators that utilize either liquid or gas as a fluid medium. A third category of fluid-mechanical exciters utilizes electric modulation of fluid properties such as electroviscosity.

Electromechanical exciters can be divided into two categories depending upon whether they derive their energy from an electric field or from a magnetic field. Each of these categories in turn can be separated into concepts involving gross motions of the transducer parts and concepts involving deformation of the transducer material.

Table I

Family Table of Exciter Concepts



5. CONCEPT EVALUATION

The exciter concepts shown in Table I were evaluated according to their capability to produce the force, stroke, and power requirements stated in Section 3. Wherever possible, the analysis dealt with the basic performance limitations of the exciters rather than evaluating specific device configurations. The final selection of the most promising exciter concept was based upon a comparative systems evaluation of the feasible concepts.

a. Driver Concepts

The following subsections qualitatively summarize the evaluation of the exciter concepts, and Table II presents these conclusions in abbreviated form. The various system configurations and the analytical work leading to their evaluation are given in Appendices B through K.

Pneumatic Piston Actuator

A large fraction of the input energy to a pneumatic system is wasted in the process of charging the actuator chamber up to operating pressure. Consequently, systems using a working fluid with a low bulk modulus are inefficient. The efficiency of a representative pneumatic actuator is calculated in Appendix B.

Hydraulic Piston Actuator Using Externally-Driven Valving

A hydraulic piston type of actuator controlled by externally-driven valving is an efficient, feasible way to excite the drill rod. Trapped fluid volumes and fluid line lengths must be minimized in order to achieve the high excitation frequency.

One way to implement the valving is to use a rotary valve as shown in Figure C-1 of Appendix C. A block diagram of this type of system is shown in Figure 1. The performance of this type of exciter is predicted in Appendix C. Due to the large working pressures that are available (e.g., 2500 psi), this type of system can be made extremely compact.

Hydraulic Piston Actuator Controlled by a Gear Pump Pressure Pulse Generator

A hydraulic piston actuator can be controlled by pressure pulses that are generated by a modified gear pump which combines the functions of pumping and valving. Although this technique is feasible, considerable development would be required to eliminate

Table II

Summary of Concept Evaluation

Fluid-Mechanical Concepts

Concept	Major Limitation	Status
Pneumatic Piston Actuator	large compressibility power loss	not acceptable
Hydraulic Piston Actuator, Externally Driven Valving	frequency response	feasible, moderate development required
Hydraulic Piston Actuator, Gear Pump Pressure Pulse Generator	frequency response, erratic behavior due to localized geometric irregularities	feasible, but requires considerable development
Unstable Fluid Bearing Actuator	low efficiency since a large fraction of the flow does no useful work	not acceptable
Hydraulic Piston Actuator Driven by a Fluidic Oscillator	possible low efficiency, excessive jet switching times, and tuning problems	feasibility is questionable, requires considerable development
Hydraulic Piston Actuator Driven by a Hydrodynamic Oscillator	low efficiency	not acceptable
Piston Actuator Driven by a Two Phase Fluid	low efficiency, impractical	not feasible
Electroviscous Fluid Actuator	low force and efficiency	not feasible
Magnetic Fluid Actuator	low force and efficiency	not feasible

Table II

Summary of Concept Evaluation

(Continued)

Electromechanical Concepts

Concept	Major Limitation	Status
Piezoelectric Exciter	durability, heat dissipation, high voltage required	feasible, moderate development required
Electrostatic Exciter	very low force per unit area	not feasible
Magnetostrictive Exciter	low efficiency, heat dissipation	feasible, but inferior to piezoelectric
Electromagnetic Exciter	low force per unit area, large weight	not feasible

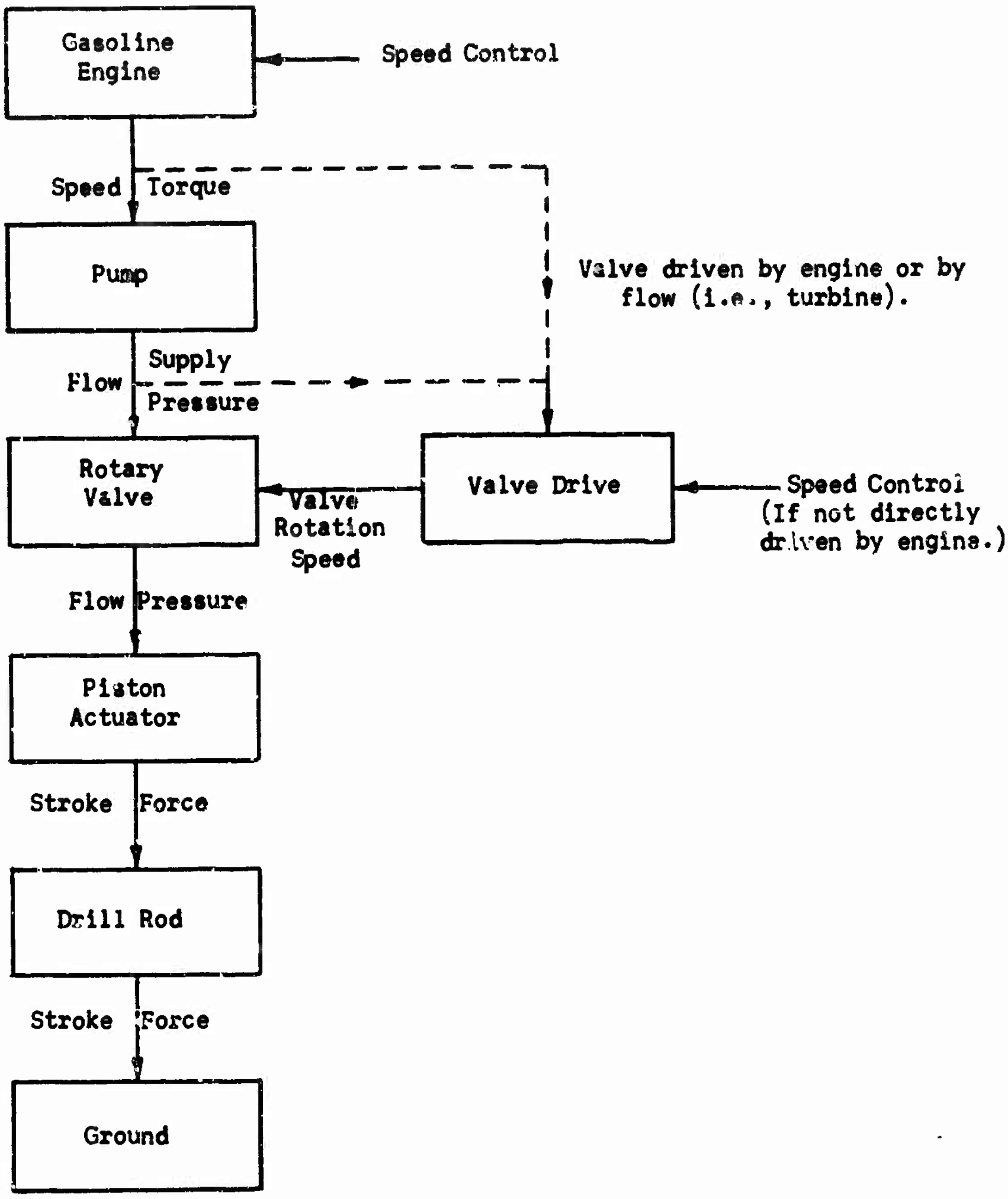


Figure 1. Block Diagram of Externally-Valved Hydraulic Exciter

excessively high instantaneous pressures or erratic behavior due to wear in the pump bearings and other localized geometric irregularities.

A preliminary study of this type of device is given in Appendix D.

Unstable Fluid Bearing Actuators

The oscillation of an unstable bearing can be used to excite the drill rod. Appendix E shows that the efficiency of this type of device is unacceptably low because most of the flow that must be supplied to the bearing does not perform useful work.

Hydraulic Piston Actuator Driven by a Hydraulic Fluidic Oscillator

As discussed in Appendix F, the feasibility of a fluidically powered piston actuator is questionable due to problems of efficiency, frequency response, and frequency tuning. Development of this device would require considerable advances in the state-of-the-art.

Hydraulic Piston Controlled by a Hydrodynamic Oscillator

Several hydrodynamic oscillators using pressure wave effects have been reported in patents and in the technical literature[2, 3, 4, 5]. A preliminary investigation of these devices has indicated that their efficiencies are considerably less than 60%. One device having a 400-cps 5-hp output had a 15% efficiency.

An additional disadvantage of these devices is that it is difficult to change their oscillation frequencies to "tune" them to the rod resonance.

In view of these drawbacks, these devices were not considered further.

Piston Actuator Driven by a Two-Phase Fluid

The pressure in a piston actuator can be controlled by the collapse and formation of vapor bubbles in a two-phase fluid. Appendix G shows that this technique has an undesirably low efficiency.

Electroviscous Fluid Actuators

Transducers that utilize fluids whose viscosity can be electrically modulated are still in an early stage of development. The force levels in existing devices are far lower than the present application requires, and this type of device has a low efficiency. Consequently, at the present time, electroviscous exciters are not feasible for this application.

Magnetic Particle Fluid Actuators

Transducers that utilize fluids with suspensions of magnetic particles are still in the experimental stage. Due to magnetic saturation effects, the force per unit area will be unacceptably low for the present application.

Piezoelectric Exciter

It is shown in Appendix H that the use of a piezoelectric exciter is a feasible means of exciting the drill rod. Possible problem areas are transducer breakage and heat dissipation. In addition, there is a safety hazard arising from the required high-voltage power supply (e.g., 1000 volts). A block diagram of a piezoelectric exciter system is shown in Figure 2.

Magnetostrictive Exciter

Magnetostrictive transducers, discussed in Appendix I, appear to be a feasible means of exciting the drill rod. However, literature in the field of high-power ultrasonics indicates that piezoelectric systems predominate over magnetostrictive systems. The drill rod exciter application requires five times more power than existing magnetostrictive systems produce. Since magnetostrictive transducers are less than half as efficient as comparable piezoelectric transducers, they were rejected in favor of the piezoelectric devices.

Electromagnetic Exciter

Due to magnetic saturation effects, electromagnetic devices have low working force per unit area. It is shown in Appendix J that the resulting large size and weight of this type of exciter makes it unsuitable for the high-frequency drill rod exciter application.

Electrostatic Exciter

Appendix K shows the force levels that can be achieved with this type of device are unacceptably low.

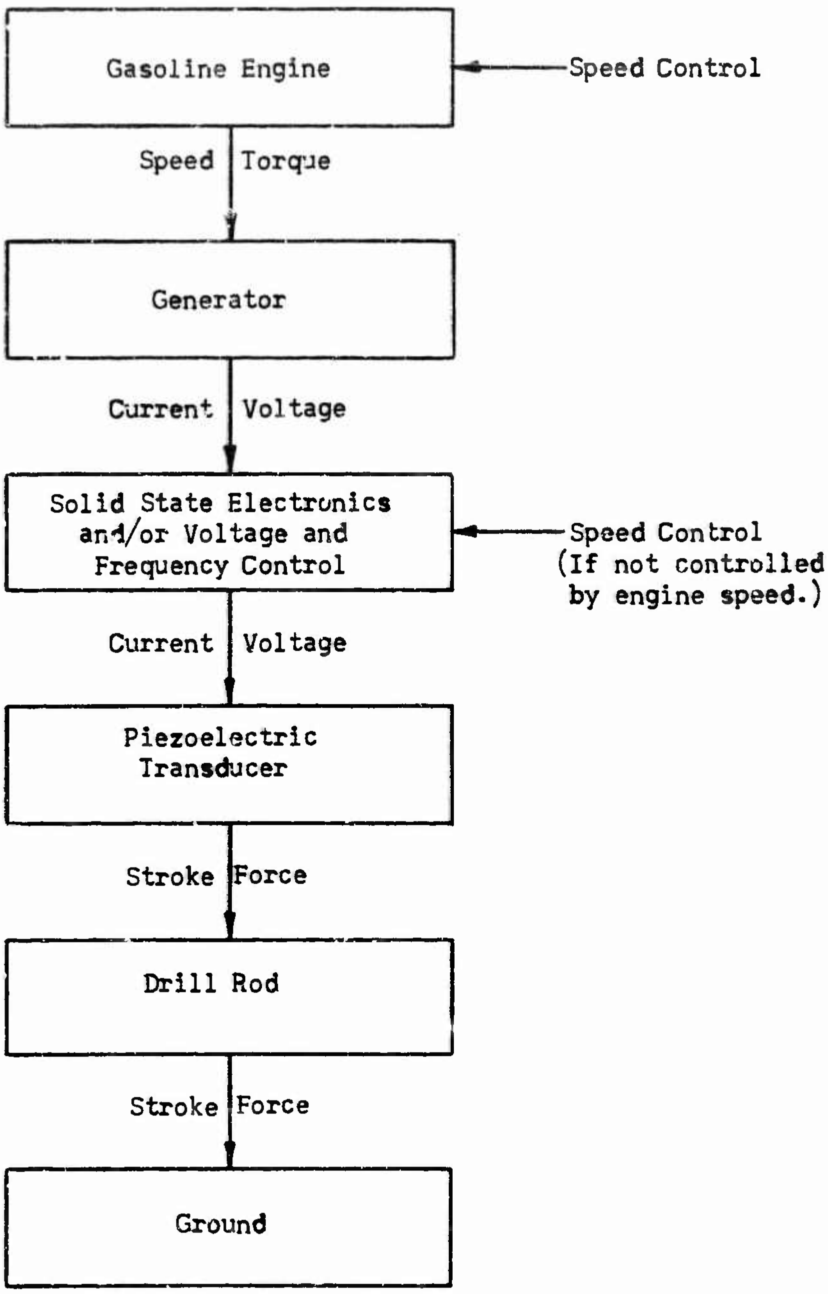


Figure 2. Block Diagram of Piezoelectric Exciter System

Summary of Exciter Concept Evaluation Conclusions

The following four concepts were found to be feasible ways to excite the drill rod:

hydraulic actuator controlled by externally-driven valving;

hydraulic actuator controlled by pressure pulse generator;

piezoelectric transducer; and

magnetostrictive transducer.

The pressure pulse generator system was not considered further because it requires considerably more development work than the externally-valved hydraulic system. The magnetostrictive system was not considered further because it is considerably less efficient than the piezoelectric system. The two remaining systems, hydraulic and piezoelectric, are compared in terms of their overall system characteristics in the following section.

b. Exciter System Evaluation

Analysis has indicated that the hydraulic actuator employing a rotary valve and the piezoelectric transducer are superior to the other exciter concepts that were investigated. Consequently, these two systems were examined further regarding their attendant power supplies and their cost, weight, safety, and other important characteristics. This section presents a comparison between the two systems. The information is summarized in Tables III and IV, and the auxiliary equipment is described in more detail in Appendices L and M. A detailed discussion of the two concepts is given in Appendices C and H, and block diagrams of the complete systems are shown in Figures 1 and 2, respectively.

Table III

Comparison between Hydraulic and Piezoelectric Systems

	Factor	Superior System	Comments
1.	Weight*	Hydraulic	The hydraulic system is 20 lbs lighter than the lightest piezoelectric system.

*See Table IV.

Table III

Comparison between Hydraulic and Piezoelectric Systems (Cont'd)

	Factor	Superior System	Comments
2.	Safety	Hydraulic	The piezoelectric system has a high-voltage shock hazard.
3.	Durability	Hydraulic	The piezoelectric ceramic and the electrical circuitry are more permanently damaged by rough handling than the hydraulic components.
4.	Reliability	Hydraulic	The system requirements can be met with standard off-the-shelf hydraulic components whose reliability has been proven. A piezoelectric transducer in the 5-hp range represents the current state-of-the-art. Its reliability is unknown.
5.	Cost*	Hydraulic	The hydraulic system is 1/3 the cost of the least expensive piezoelectric system and 1/7 the cost of the lightest piezoelectric system.
6.	Self-Induced Destruction	Hydraulic	If the cooling fan or the voltage control circuitry fails, the piezoelectric transducer will destroy itself. No such danger exists with the hydraulic system.
7.	Self-Contained System	Hydraulic	The hydraulic system can be self-contained. The possibility of building a self-contained piezoelectric system is marginal due to the relatively large total system weight.

*See Table IV.

Table III

Comparison between Hydraulic and Piezoelectric Systems (Cont'd)

	Factor	Superior System	Comments
8.	Maintenance	Comparable	The hydraulic system has more moving parts than the piezoelectric system, but they are self-lubricating.
9.	Mobility of Transducer	Piezo-electric	Assuming that neither system is completely self-contained, a transducer connected to a stationary power supply by an electric cable is more mobile than one that is connected by two hydraulic hoses.
10.	Frequency Tuning	Comparable	The hydraulic system will be easier to accurately tune than an engine-driven high-frequency generator and more difficult to tune than a solid-state electrical oscillator. The hydraulic system will have to be tuned more frequently due to temperature sensitive viscosity effects.
11.	Multiple Transducers Driven by a Single Power Supply	Piezo-electric	Several piezoelectric transducers could be powered by one electrical power supply. The relatively short allowable length of hydraulic hoses makes this less desirable for a hydraulic system.
12.	Cooling	Hydraulic	The piezoelectric system requires forced convection, whereas the hydraulic system dissipates heat by free convection and radiation.

Table IV

Various System Configurations Indicating Estimated Costs and Weights

Hydraulic Transducer		Piezoelectric Transducer				
Drill Rod 10 lb		Drill Rod 10 lb				
Hydraulic Piston, Disk Valve, Speed Control 4 lb $200	Hydraulic Piston, Disk Valve, Speed Control 4 lb $200	Piezoelectric Transducer, Cooling Fan 10 lb $500-$1000	Piezoelectric Transducer, Cooling Fan 10 lb $500-$1000	Piezoelectric Transducer, Cooling Fan 10 lb $500-$1000	Piezoelectric Transducer, Cooling Fan 10 lb $500-$1000	Piezoelectric Transducer, Cooling Fan 10 lb $500-$1000
Gear Pump 3 lb $35	Gear Pump 3 lb $35	Solid State 4 kva Driver 50 lb $500	Solid State 4 kva Driver 50 lb $500	Solid State 4 kva Driver 50 lb $500	Step up Transformer 15 lb $ 20	Voltage and Speed Control 15 lb $200
Mounting Structure, Pressure Relief Valve 10 lb $50	Hydraulic Hoses 10 lb	Transformer and Rectifiers 10 lb $ 50		Transformer and Rectifiers 10 lb $ 50	Voltage and Speed Control 15 lb $200	Mounting Structure 10 lb $ 50
Reservoir, Oil, Pressure Relief Valve 28 lb $50	Reservoir, Oil, Pressure Relief Valve 28 lb $50	Engine-Generator Set 5 kva, 115 V 60 cps 230 lb $500	Engine-Generator Set 5 kva, 200 VDC 300 lb $900	Generator 4.5 kva 30VDC (7200 rpm) 25 lb $1000	Generator 3 kva, 115V 1225 cps at 10,500 rpm 50 lb $400	Custom Designed Generator 1600 cps at 12,000 rpm 4 kva, 1000 V 25 lb $2000
Gasoline Engine $100 10 lb-40 lb	Gasoline Engine $100 10 lb-40 lb			Gasoline Engine 10 lb - 40 lb $100	Gasoline Engine, Gear Train Speed Control $200 [illegible] lb - 45 lb	Gasoline Engine, Gear Train Control 15 lb - 45 lb $200
(65 lb - 95 lb $450)	(65 lb - 95 lb $450)	(310 lb $1550 - $2050)	(370 lb $1990-$2400)	(115 lb - 145 lb $2150 - $2650)	(115 lb - 145 lb $1350 - $1850)	(85 lb - 115 lb $2950 - $3450)
Self-Contained System	Systems with Separate Power Supplies					Self-Contained System

- - - Indicates interface between components on the drill rod and stationary power pack.

The major factors that were compared are discussed briefly in the following subsections:

Weight

Table IV indicates the weights of different hydraulic and piezoelectric systems. A piezoelectric system utilizing a special high-speed, light-weight generator is 20 to 50 lbs heavier than a hydraulic system. If a standard engine generator set is used, the piezoelectric system is several hundred pounds heavier than a hydraulic system.

The hydraulic and the piezoelectric transducers, taken separately from their power supplies, have comparable weights. However, the piezoelectric transducer requires a cooling fan. Consequently, the minimum weight that must be mounted on the drill rod is less for a hydraulic than for a piezoelectric exciter.

Safety

The high voltage used in the piezoelectric system is an inherent safety hazard. Faulty cable connections or transducer breakage could result in a short circuit that could electrocute the operator. This danger is greatly enhanced by the ice and snow that will be present in the arctic environment.

A hydraulic pressure of several thousand psi is not a significant safety hazard since oil is relatively incompressible and therefore has a low energy density. A properly designed system that is provided with adequate heat transfer area will operate at safe oil temperature levels.

Durability

The piezoelectric transducer comprises a series of brittle ceramic wafers. If the transducer is subjected to large shock loads, or the strain amplitude is allowed to exceed a critical value, the ceramic will fracture and the device will be destroyed. Rough handling can damage the electrical circuitry associated with the transducer. Excessive temperatures in the transducer will destroy the piezoelectric properties of the ceramic.

The hydraulic system can be constructed from rugged materials that are not subject to the above types of failure.

Reliability

As discussed in the section on Self-Induced Destruction, page 18, the piezoelectric transducer is inherently subject to catastrophic failure resulting from failure of its voltage control or cooling systems. It is also, by nature of its mechanical properties, highly susceptible to damage from rough handling. Moreover, at the power levels required for this application, it represents the current state-of-the-art and its reliability and life under prescribed operating conditions is not known.

The hydraulic transducer, on the other hand, is not susceptible to any of these types of failure, and is known to be highly reliable when good design practice is employed.

The same comparison holds for the various elements in the power train. In general, the electrical and electronic components represent the present state-of-the-art at the high power levels involved and are known to be inherently less reliable than their hydraulic counterparts.

Cost

Table IV indicates the estimated costs of various hydraulic and piezoelectric systems. The cost of the hydraulic system will be approximately one-third that of the least expensive piezoelectric system. Comparison between the embodiment of each concept which could be self-contained on the drill rod shows that the piezoelectric system would be at least seven times as costly.

Self-Induced Destruction

If the excitation voltage control circuit malfunctions while the piezoelectric transducer is not properly loaded, the resulting strain amplitude will destroy the transducer. If, for some reason, the forced convection cooling is impaired, the resulting temperature rise in the transducer will reduce its efficiency causing increased internal heat generation and catastrophic failure. There is no such inherent catastrophic failure possible for the hydraulic transducer.

Self-Contained System

Table IV indicates the total weights of different combinations of components for the various systems. Depending on the

type of gasoline engine that is used, the hydraulic system will weigh between 65 and 95 lbs, and the lightest type of piezoelectric system will weigh between 85 and 115 lbs. If it is assumed that the maximum allowable portable system weight is 85 lbs, then mounting all of the piezoelectric components in a self-contained unit is only marginally feasible. Any unanticipated extra weight would cause the total weight to exceed the limit. The hydraulic system can likely be mounted on the drill rod in a self-contained system even if some additional components are necessary. If the maximum allowable weight is 75 lbs, then only the hydraulic system can be a self-contained unit.

Maintenance

Both the piezoelectric and hydraulic systems require the maintenance of a gasoline engine. In addition, the hydraulic system will require seal replacement and bearing lubrication, while the piezoelectric system requires only generator bearing lubrication. Whereas failures in a piezoelectric system are infrequent, their diagnosis can be complicated and will require auxiliary electronic equipment manned by skilled personnel. A failure in the hydraulic system can be easily diagnosed and replacement of parts will be a simple operation.

Transducer Mobility and Multiple Transducers

Under certain conditions, it might be desirable to separate the power supply from the drill rod. For example, it might be desired to operate a number of separate drills from a large central power supply.

For this system configuration, the piezoelectric system would be superior to the hydraulic system since the electric cables can be made lighter, more flexible, and have less dissipation than the hydraulic hoses.

Frequency Tuning

The frequency of the hydraulic system can be varied by adjusting the flow in the turbine that drives the rotary disk valve as described in Appendix C. Fluid viscosity changes will require that this be frequently done, especially during the warm-up transient.

If the piezoelectric transducer is directly driven by a high-frequency generator, frequency tuning is accomplished by controlling the speed of the gasoline engine. Frequency tuning can

also be performed electrically if a high-power, solid-state oscillator is used to drive the transducer.

Controlling the turbine speed will be easier than controlling the gasoline engine speed, and more difficult than controlling the electrical oscillator frequency. Thus, depending on the combination of components used, either system could display superiority in this respect.

Cooling

The piezoelectric system requires a motor-driven fan to provide forced convection.

The hydraulic fluid transports heat away from the transducer to a reservoir where the heat is dissipated by free convection and radiation. Thus, the hydraulic system is more easily and more reliably cooled than the piezoelectric system.

c. Results of Concept Evaluation

The detailed comparison between the hydraulic and piezoelectric drilling systems indicates that the hydraulic system is superior to the piezoelectric system in practically every system category that was investigated. Consequently, the externally-driven rotary valve hydraulic system was selected for development as a laboratory model.

6. EXPERIMENTAL TEST PROGRAM

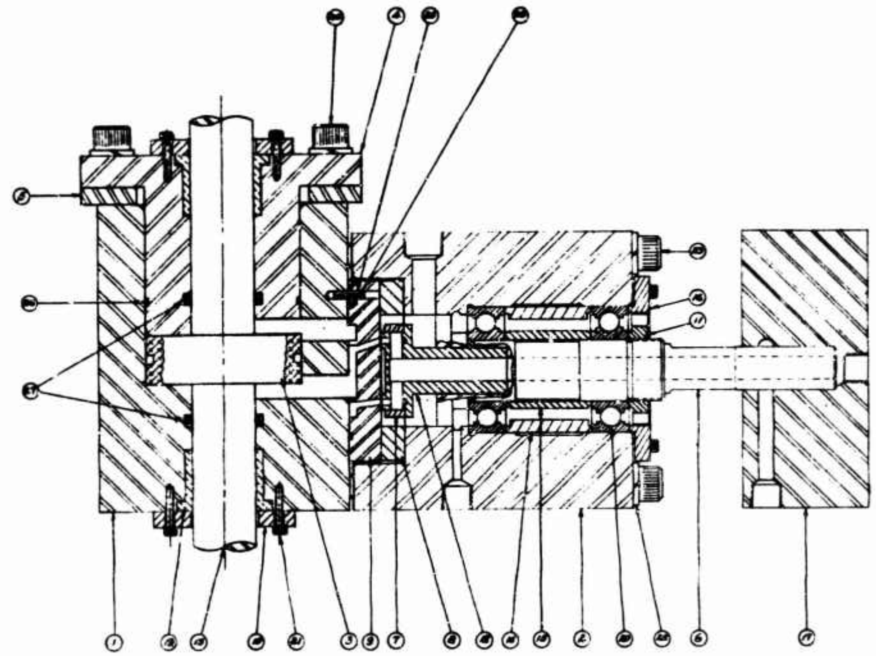
The work performed in the first two phases indicated that the externally-driven rotary valve hydraulic actuator was the most promising system. In the third phase, a laboratory model was developed in order to demonstrate that this type of exciter could produce and maintain a resonant vibration in a drill rod. A power level of 3 hp was selected for this purpose. No attempt was made to optimize the design by minimizing size, weight, or production cost.

a. Description of the Laboratory Model Hydraulic Drill Rod Exciter

An assembly drawing of the laboratory model exciter is shown in Figure 3. The exciter consists of two major subassemblies: the vibrating piston (Parts 1, 3, 4, and 5); and the externally-driven, rotary valve (Parts 2, 6, 7, 8, and 9). Figures 4 and 5 are photographs of the subassemblies.

Assembly of Hydraulic Drill Rod Exciter

Figure 3



REQ'D	ITEM	PART NO.	ITEM NO.
1	PARKER "O" RING BUNA "N"	2-39	28
2	PARKER "O" RING BUNA "N"	2-216	27
7	PARKER "O" RING BUNA "N"	2-036	26
1	[illegible] ROLL PIN		25
6	[illegible] SOCK. HD CAP SCR.		24
6	[illegible] SOCK. HD CAP SCR.		23
[illegible]	[illegible] STD. LOCK WASHER		22
14	5-40 x [illegible] SOCK. HD CAP SCR.		21
2	BARDEN BEARING [illegible]	205 FF	20
2	CAST-BRONZ BUSHING	FB-1620-8	19
1	BRACKET, DRIVE MOTOR	B-QM6740-18	18
1	HOUSING, PRESSURE FEED	A-QM6740-17	17
1	RETAINER RING, OUTER RACE	A-QM6740-16	16
1	SPACER, INNER	A-QM6740-15	15
1	SPACER, OUTER	A-QM6740-14	14
1	ROD, DRILL	A-QM6740-13	13
1	VALVE DISK	A-QM6740-12	12
1	RETAINER, INNER RACE	A-QM6740-11	11
2	RETAINER, BUSHING	A-QM6740-10	10
[illegible]	VALVE MANIFOLD PLATE	A-QM6740-9	9
1	VALVE SPACER PLATE	A-QM6740-8	8
1	DISK, VALVE RING	A-QM6740-7	7
1	DRIVE SHAFT	B-QM6740-6	6
1	PISTON SPACER BLOCK	B-QM6740-5	5
1	CYLINDER HEAD	A-QM6740-4	4
1	PISTON	B-QM6740-3	3
1	VALVE HOUSING	B-QM6740-2	2
1	PISTON HOUSING	C-QM6740-1	1

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ASSEMBLY

4-15-60 — D — QM6740-19

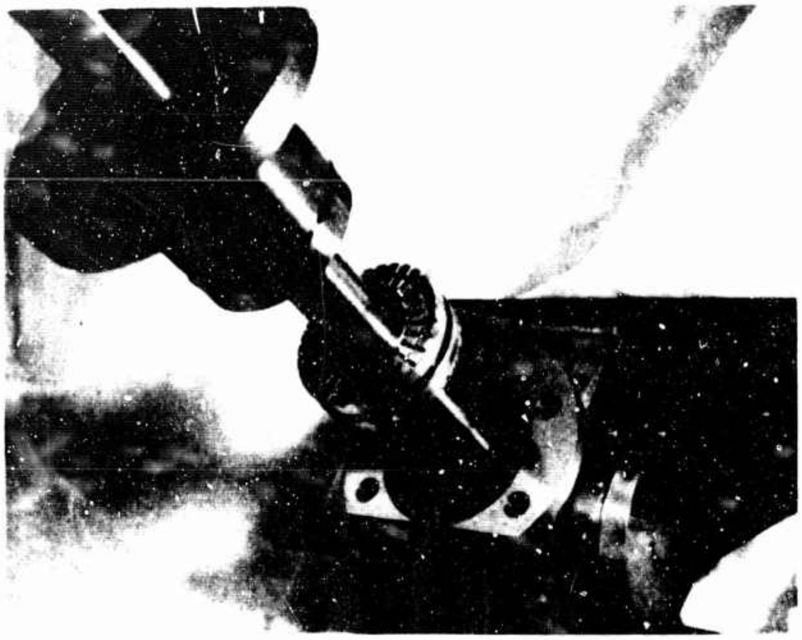

Figure 4. Photograph of Piston Subassembly

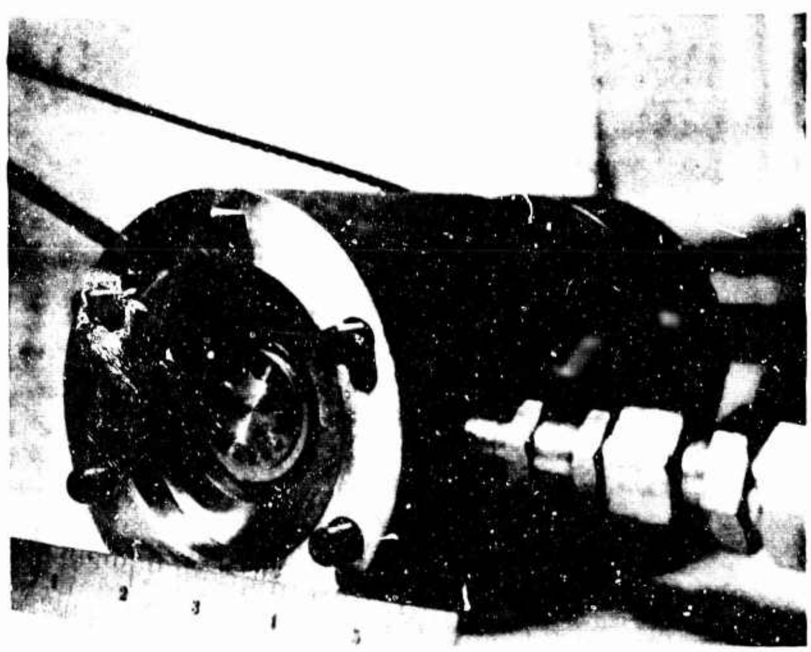

Figure 5. Photograph of Valve Subassembly

A series of fluid flow ports is provided in the hollow drive shaft, rotary valve, and manifold plate to communicate alternately with each side of the piston. The piston stroke can be varied by changing the thickness of the spacer block, Part 5. The drill rod is laterally supported by bushings, Part 19, in the piston housing, Part 1. The piston is rigidly attached to the drill rod.

Hydraulic fluid from the pressure feed housing, Part 17, flows through the hollow drive shaft, Part 6, to the rotary valve. Ports are arranged in the valve and manifold plate, Part 9, so that each side of the piston is alternately connected to fluid supply and exhaust. The axial clearance between the rotating valve and the manifold plate is determined by the spacer, Part 8.

No special design was developed for the disk valve to minimize unbalanced pressure loading. This configuration is described in detail in Appendix N.

All parts of the laboratory model were made of steel except the aluminum piston and sintered bronze bushings. The drive shaft was driven by a universal motor through a positive drive belt and pulleys. Motor speed was adjusted by an a.c. variable transformer to tune the frequency of excitation to the drill rod resonant frequency.

b. <u>Design of the Laboratory Model Hydraulic Drill Rod Exciter</u>

A nominal power level of 3 hp was selected for the laboratory model exciter. The drill rod was 72 inches long, and the piston was mounted three-quarters of an inch (axially) from the center of the rod. The required driving force and stroke were determined from Figure A-3 in Appendix A. The pertinent design parameters are summarized in Table V, and the estimated performance parameters and design goals are summarized in Table VI.

The design of the passages in the manifold plate and in the piston housing is a compromise between minimum volume for fast response, and large area to keep pressure losses to a minimum. In particular, large pressure losses due to fluid accelerations must be avoided. Another factor that contributed to the total trapped volume was the radial and circumferential grooves that had to be machined into the piston faces to reduce squeeze film damping to an acceptable level.

It should be emphasized that a final design utilizing a turbine drive and fluid bearings would be far more compact. With additional engineering effort, the housing sizes connecting line lengths and rotating clearances could all be reduced. This would

Table V

Hydraulic Exciter Laboratory Model Design Parameters

1. Drill Rod

Length	=	6 ft
Excitation Frequency	=	1425 cps
Rod Diameter	=	1 inch
Material	–	steel

2. Actuator

Area	=	4 in^2
Piston Diameter	=	2.5 inches
Stroke	=	$\pm$ 0.0015 inches

Position on drill rod, 3/4 inch below center of the rod

Supply Pressure	=	1000 psi
Grooves	–	30 radial and 2 circumferential 1/16 x 1/16 grooves on the piston faces

3. Disk Valve

Holes	–	16 holes, 0.1 inch diameter on 1 inch pitch circle
Clearance between disk valve and manifold plate	–	0.001 inch

4. Connecting Lines

Area	=	0.2 in^2
Length	=	1.5 inches

5. Trapped Volume

Volume	$\approx$	0.6 in^3 for each piston chamber

<u>Table VI</u>

<u>Design Goals and Estimated Performance Parameters</u>

1. <u>Nominal Exciter Output</u>

Maximum force	=	$\pm$ 4000 lb
Maximum stroke	=	$\pm$ 0.0015 in
Power	=	3 hp

2. <u>Flows</u>

Actuator	=	8 gpm
Disk valve leakage	=	1 1/4 gpm
Compressibility	=	1 1/2 gpm

3. <u>Pressure Drops</u>

Inertia pressure drop per line	=	250 psi – 90° out of phase with velocity
Resistive drop in disk valve	=	100 psi
Effect of piston inertia force	=	20 psi – 90° out of phase with velocity

4. <u>Power Losses</u>

Squeeze film damping	=	1/2 hp at 150°F oil temperature occurring between the piston forces and the ram end walls

5. <u>Dynamic Response</u>

R C cutoff frequency	=	3700 cps	(see Appendix C)
L C surge frequency	=	3500 cps	(see Appendix C)

decrease leakage flows and line pressure drops due to fluid inertia and enable a more pronounced resonance to be produced.

c. <u>Performance of Laboratory Model Hydraulic Drill Rod Exciter</u>

The laboratory model drill rod exciter exhibited a resonant vibration at approximate 1415 cps. From Equation A-1, Appendix A, the predicted resonant frequency was 1475. (Due to a machining error the rod length had to be reduced to 70 inches.) The deviation of the actual and predicted resonant frequencies is probably due to the effect of the piston mass. The resonance was pronounced over a region of approximately 25 cps. The peak-to-peak tip vibration amplitude was 0.004 inches. This was approximately four times larger at resonance than at adjacent frequencies.

It is felt that fluid inertia pressure drops in the supply and exhaust lines caused the excitation to be considerably less than the design goals. It is anticipated that if these drops were reduced by utilizing a more compact valve-actuator design, the vibration amplitude would be significantly increased.

Preliminary tests were made of the performance of the system drilling frozen sand. Although quantitative measurements of drilling rates were not obtained, the results indicated that an experimental study of the interaction between the drill rod and the permafrost should be made as part of future work on a vibratory drilling system. One specific area that should be investigated is the optimum design shape of the drill tip so that there is adequate provision for clearing of the displaced material.

7. <u>CONCLUSIONS</u>

The primary conclusion made from this analytical and experimental study is that the most promising way to excite the drill rod is to use some form of hydraulic excitation. This conclusion is based upon an examination of concept feasibility and excitation system characteristics.

Additional conclusions are as follows:

a. The overall process analysis showed that a power level of the order of 5 hp will be required to meet the drilling rate criterion. This will require a typical exciter output having a peak force of 5000 lbs and a peak amplitude of 3 mils.

b. The most promising hydraulic exciter concept capable

of meeting the above criteria and satisfying the durability and weight requirements is a hydraulic piston driven by a rotary disk valve.

c. A hydraulic piston driven by gear pump mesh pressure transients also appears potentially feasible but would require a substantial development effort to establish final feasibility and produce a practical field system.

d. Piezoelectric and magnetostrictive transducers are feasible means of excitation but were found less promising than the hydraulic system on the basis of overall system size, weight, cost, and durability.

e. Other exciter concepts including pneumatic pistons, unstable bearings, two-phase fluid systems, electroviscous fluids, magnetic fluids, electromagnetic transducers, and electrostatic transducers were found unfeasible for the present design criteria. In general the electromechanical concepts were not capable of sufficient force within practical size and weight limitations. The remaining systems were found to have prohibitive power losses.

f. It has been experimentally demonstrated that a rotary disk valve hydraulic actuator can excite a 6-ft long, 1-inch diameter rod into a longitudinal resonant vibration.

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APPENDIX A

DRILL ROD EXCITER OUTPUT PARAMETERS

In this appendix, the important output parameters for the drill rod exciter are established as a function of the several system variables. Representative values of these quantities were used to evaluate the feasibility of the exciter concepts that were studied.

1. Excitation Frequency

To excite the drill rod into axial resonance by applying a vibratory force at its midpoint, the excitation frequency must be equal to the drill rod natural frequency, f_N.

$$f_N = \frac{1}{2} \frac{c}{\ell} \tag{A-1}$$

where $c = \sqrt{\frac{E}{\rho}}$

ℓ = drill rod length

c = velocity of sound in the drill rod

E = Young's modulus of the drill rod

ρ = density of the drill rod

Evaluation of Equation (A-1) for a 72-inch long drill rod mode of steel or aluminum indicates that the excitation frequency will be 1425 or 1390 cps, respectively.

2. Power Required to Penetrate Permafrost[6]

The power, $\mathbb{P}_p$, that is required to penetrate permafrost can be estimated as

$$\mathbb{P}_p = F_p \; V_{avg} \tag{A-2}$$

where F_p = force required to penetrate permafrost $= \frac{\sigma_c \; A}{(B-B')}$

σ_c = unconfined compressive strength of the permafrost

(B-B') = degree of compaction of the permafrost expressed as a change in the void fraction

A = area of drill rod

V_{avg} = average drilling velocity

Typical values of the permafrost parameters at 25° are

$(B-B') = 0.2$

σ_c = 400 psi to 1500 psi

In order to drill a 1-inch diameter hole at a rate of 1 fps, the required power is

$\mathbb{P}_p$ = 3 hp to 11 hp

For the purpose of exciter concept evaluation, an exciter output power level of 5 hp was chosen.

3. Exciter Force and Stroke Requirements

The size of the drilling system can be minimized if optimum power transfer is achieved by matching the output impedance of the exciter to the input impedance of the drill rod. This section summarizes an analysis of the input impedance of a drill rod that is excited in its free-free resonant mode. The required exciter maximum force and stroke as determined by this analysis are presented in graphical form.

In order to provide a tractable analysis of the drill rod input impedance, the following assumptions were made:

(a) The two sections of the drill rod above and below the exciter are separately modeled by an equivalent lumped mass and lumped spring constant. This is justified because the drill rod is assumed to be excited approximately in its first resonant mode of vibration. This mode has a node at the center of the bar and has maximum amplitudes at the ends of the bar.

(b) When the permafrost is in contact with the drill rod, it exerts a constant force, F_p that is dependent on the soil compressive strength.

(c) The interaction between the rod and the permafrost can be modeled by an equivalent dashpot having a constant, b. The value of b is determined by equating the power delivered to the permafrost to the power that is dissipated in a dashpot subjected to a sinusoidal velocity.

(d) Contact between the rod and the permafrost is intermittent, and only occurs when the sinusoidal displacement amplitude of the bar tip is near its positive peak value. The contact time is called δT.

The distributed system mode shape, together with its lumped equivalent system are shown in Figure A-1. The waveform of the drill rod tip is shown in Figure A-2.

Since the average acceleration of the drill rod into the ground is negligible, the net downward impulse must equal the net upward impulse so that the contact time, δT, is

$$F_p \, \delta T = F_{avg} \, T \tag{A-3}$$

where F_{avg} = average drilling force

T = period of vibration $\left(= \frac{1}{f_N}\right)$

From Figure A-2, the tip penetration per cycle, Δu_2, is given by

$$\Delta u_2 = u_{2_m} \left(1 - \cos 2\pi \frac{\delta T}{T}\right) \approx u_{2_m} \frac{\left[2\pi \left(\frac{\delta T}{T}\right)\right]^2}{2} \tag{A-4}$$

The average drilling velocity, V_{avg} is

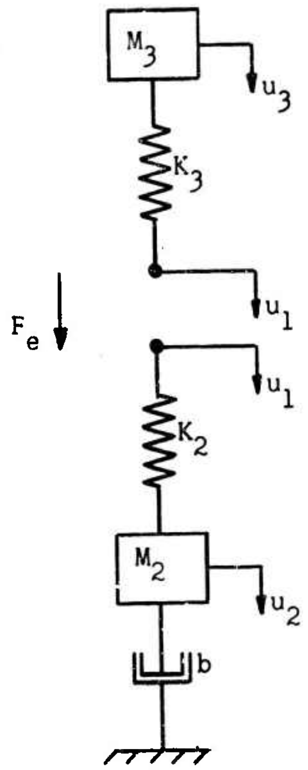
$$V_{avg} = \frac{\Delta u_2}{T} \tag{A-5}$$

Combination of Equations (A-2) and (A-3) through (A-5) yields

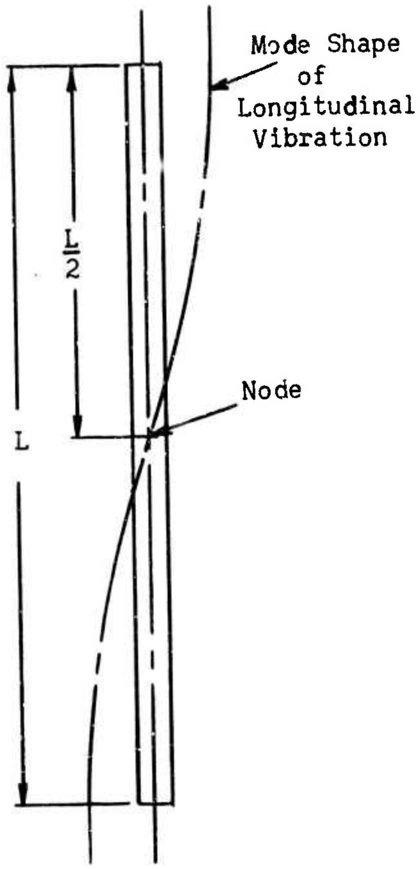
$$u_{2_m} = \frac{V_{avg}}{2 \, f_N} \left(\frac{F_p}{\pi \, F_{avg}}\right)^2 \tag{A-6}$$

The average power, $\mathbb{P}_d$, dissipated in a linear dashpot, b, subjected to a sinusoidal displacement u_2 at frequency f_N is equal to

$$\mathbb{P}_d = \frac{b}{2} \left(2\pi \, f_N\right)^2 \; u_{2_m}^2 \tag{A-7}$$



Lumped Parameter Model of Resonant Rod



Resonant Rod Showing the Free-Free Mode Shape

Figure A-1. Drill Rod Resonance Mode Shape and its Lumped Parameter Equivalent

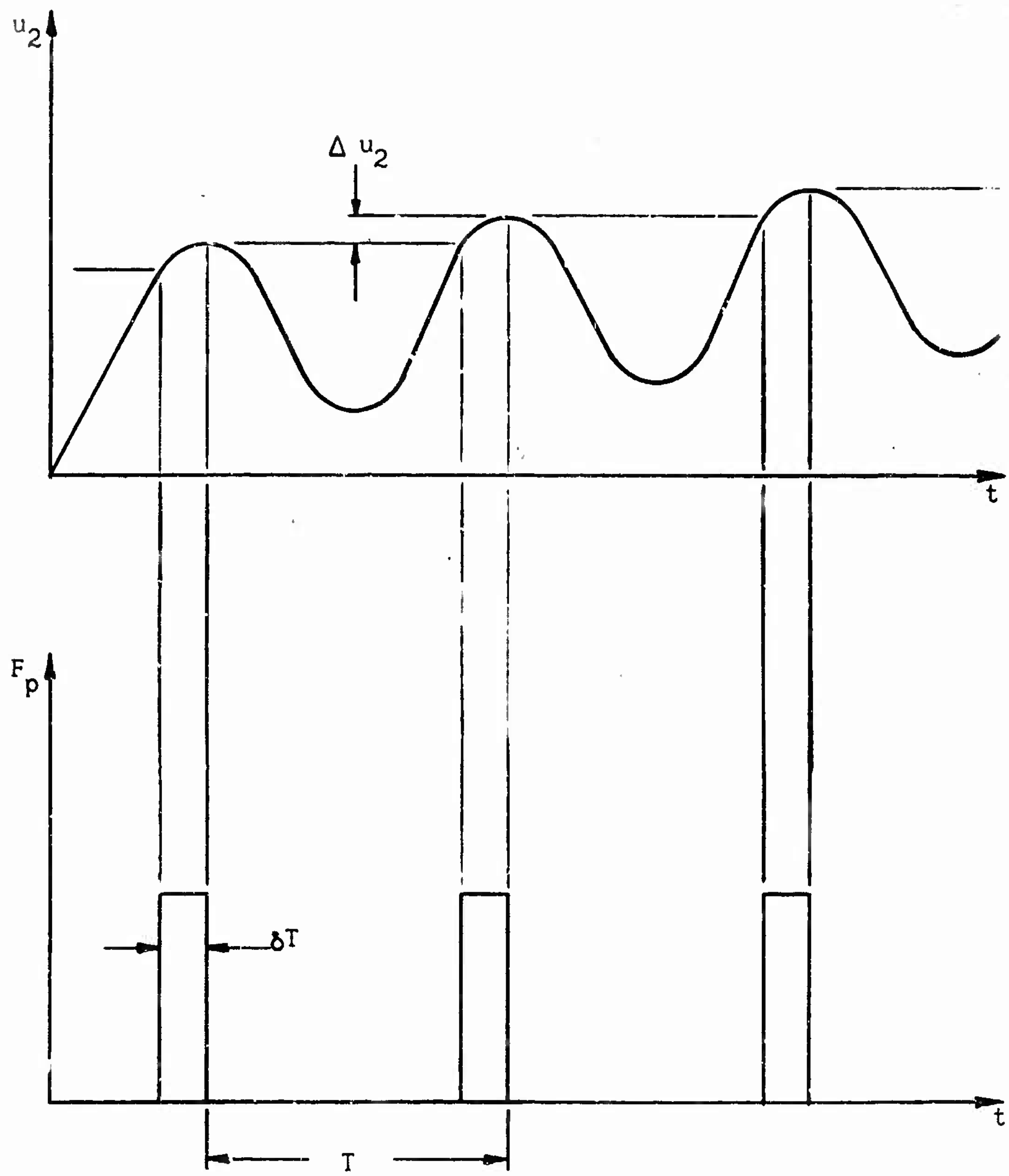


Figure A-2. Waveforms of Drill Tip Displacement and Permafrost Reaction Force

If $\mathbb{P}_d$ and $\mathbb{P}_p$ are equated, the equivalent dashpot, b, is given by

$$b = \frac{2(F_{avg})^2}{\mathbb{P}_p} \left(\frac{\pi F_{avg}}{F_p} \right)^2 \qquad \text{(A-8)}$$

Evaluation of the input impedance, Z_1 of the lumped system of Figure A-1 yields

$$Z_1 = \frac{F_e}{u_1} = \frac{(M_2 + M_3)\, S^2 \left(\frac{S^2}{\omega'^2} + 1\right) + \left(\frac{S^2}{\omega''^2} + 1\right) bS}{\left(\frac{S^2}{\omega_2^2} + \frac{b}{K_2} S + 1\right)\left(\frac{S^2}{\omega_3^2} + 1\right)} \qquad \text{(A-9)}$$

where F_e = input force at the driving point

u_1 = displacement at driving point

S = Laplace transform operator (= $j\omega$ for sinusoidal excitation at frequency ω)

$$\omega'^2 = \frac{\dfrac{K_2 K_3}{K_2 + K_3}}{\dfrac{M_2 M_3}{M_2 + M_3}}$$

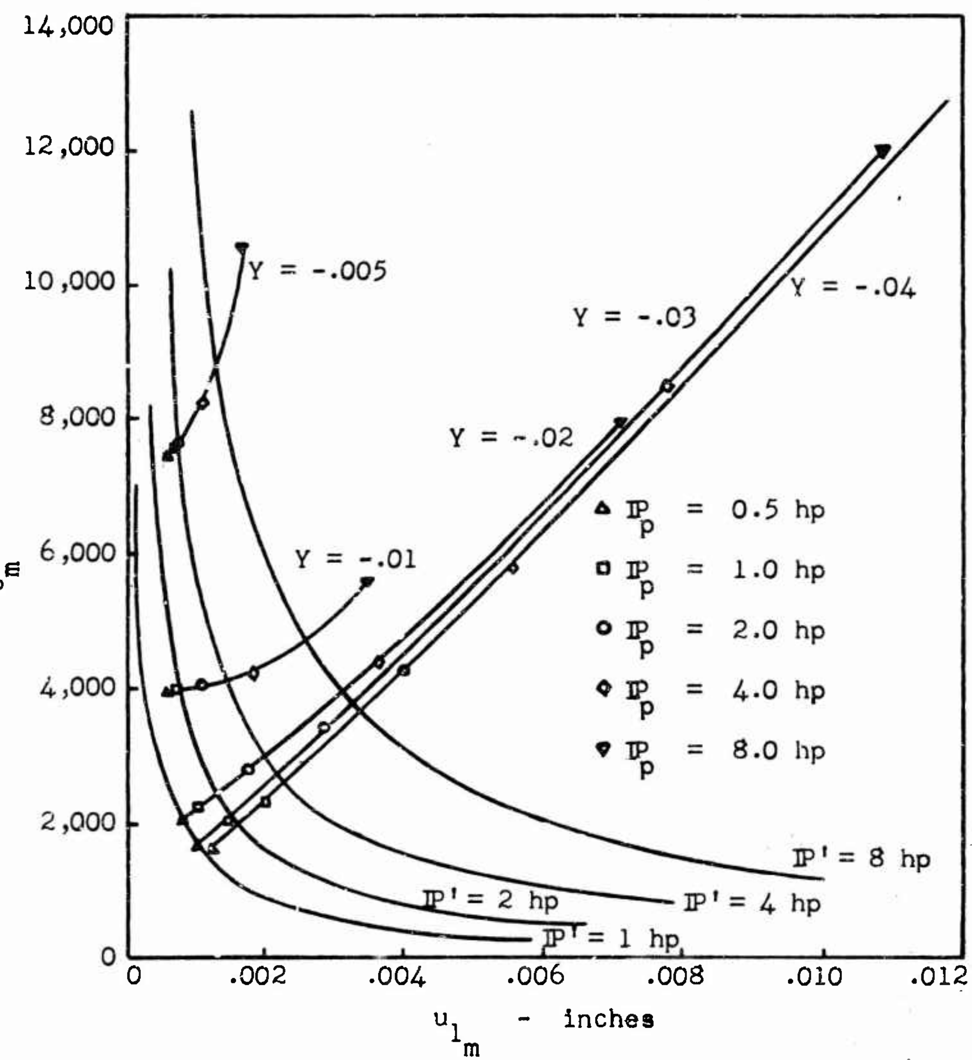
$$\omega''^2 = \frac{\dfrac{K_2 K_3}{K_2 + K_3}}{M_3}$$

The lumped spring constants and masses are functions of Y, the fractional deviation of the driving point position below the center of the bar.

Figure A-3 shows the input maximum force versus maximum stroke of the drill rod as a function of $\mathbb{P}_d$ for various values of Y for the case of

F_{avg} = 175 lb

f_N = 1350 cps



F_{avg} = 175 lb

F_p = 1500 lb

f_N = 1350 cps

L = 6 ft

A = $\pi/4$ in^2

E = 10×10^6 lb/in^2

ρg = 0.1 lb/in^3

P_p = $F_p V_{avg}$

P' = $1/2\ F_{e_m} u_{1_m}\ 2\pi f_N$

where F_p = force required to penetrate permafrost, and

V_{avg} = penetration velocity

$$b = \frac{2(F_{avg})^2}{P_p}\left(\frac{\pi F_{avg}}{F_p}\right)^2$$

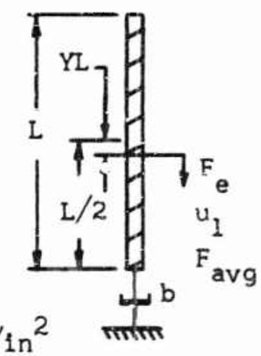


Figure A-3. Sinusoidal Force Amplitude versus Sinusoidal Displacement Amplitude with Nondimensionalized Driving Point Position, Y, as a Parameter

Superimposed on the figure are contours of constant $\mathbb{P}'$

where $$\mathbb{P}' = \frac{1}{2} F_{e_m} u_{1_m} (2\pi f_N) \tag{A-10}$$

and subscript m indicates peak amplitude

The real power delivered to the permafrost, $\mathbb{P}_p$, is given by

$$\mathbb{P}_p = \mathbb{P}' \cos \theta \tag{A-11}$$

where θ = phase shift between force and velocity at the drill rod driving point

Examination of Figure A-3 indicates that θ is smallest for small Y. Consequently, larger values of F_{e_m} and u_{1_m} (requiring a larger system) are needed to deliver a given amount of real power, $\mathbb{P}_p$, for $Y = 0.04$ than for $Y = 0.01$.

In order to match the exciter output impedance to the drill rod input impedance, the maximum values of the exciter stroke, x_m, and force, F_m are selected so that

$$x_m = u_{1_m} \tag{A-12}$$

$$F_m = F_{e_m}$$

The quantities u_{1_m} and F_{e_m} are determined from Figure A-3 once $\mathbb{P}_p$ and Y are chosen. For the purpose of concept evaluation, a Y of 0.01 and a value of 5 hp were selected so that the required exciter force and stroke are

$$F_m = \pm 5000 \text{ lb}$$

$$x_m = \pm 0.003 \text{ in}$$

The tip deflection, u_{2_m} that is required to deliver this power level to the permafrost is

$$u_{2_m} = 0.11 \text{ inches}$$

The large required drill tip deflection, u_{2_m}, is likely to cause the drill rod to experience a fatigue failure. For example, the peak dynamic

stress, σ_p, of a steel free-free rod resonating with a tip amplitude of $u_{2_m} = 0.11$ inches is

$$\sigma_p = \left(\frac{E\pi}{\ell}\right) u_{2_m} \tag{A-13}$$

$$= 140{,}000 \text{ psi}$$

where $E = 30 \times 10^6$ psi for steel

Special high fatigue strength alloys would be required to tolerate this stress level. To reduce u_{2_m} and σ_p to more desirable levels, Equation (A-6) indicates that a lower average drilling velocity must be accepted. Equation (A-2) shows that $\mathbb{P}_p$ is similarly reduced. Thus, the power that a resonating drill rod can deliver to the ground is limited by the strength of the drill rod material regardless of the characteristics of the drill rod exciter.

For example, if u_{2_m} is reduced to 0.055 in, then V_{avg} and $\mathbb{P}_p$ are reduced to half of their previous nominal values of 1 ft/sec and 5 hp.

APPENDIX B

PNEUMATIC PISTON ACTUATOR

In this appendix, the efficiency of a suitably valved pneumatic piston-type actuator was investigated. A typical arrangement is schematically shown in Figure B-1. The low bulk modulus of the fluid results in unacceptably low system efficiencies since a large amount of air flow is necessary to charge the piston chambers up to supply pressure even if the piston does not move. This compressibility flow does no useful work, and requires large power expenditures.

A lower bound on the work per cycle, W, done in charging the piston chamber from ambient pressure, p_a, up to supply pressure, p_s, is given by

$$W = 2\frac{v}{k}(p_s - p_a) \qquad \text{(B-1)}$$

where k = ratio of specific heats (1.4 for air)

v = total trapped volume of each piston chamber and its supply line

The power, $\mathbb{P}_c$ that must be supplied is

$$\mathbb{P}_c = W f_N \qquad \text{(B-2)}$$

where f_N = the operating frequency (1350 cps)

A system with a 1000 psi supply pressure requires at least a 3 in^2 actuator area in order to obtain a 5000 lb output force. For a peak to peak amplitude of 0.003 inches, the minimum actuator volume is 0.03 in^3 The orifice area must be at least 0.1 in^2 in order to pass the required chamber flow rate. A 1 inch long chamber supply line having this area results in

$$v = 0.13 \text{ in}^3 \qquad \text{(B-3)}$$

Evaluating Equation (B-2), the compressibility power loss is

$$\mathbb{P}_c = (2)\frac{(0.13 \text{ in}^3)}{1.4}\left(1000 \frac{\text{lb}}{\text{in}^2}\right)(1350 \text{ cps})\left(\frac{1}{6600 \frac{\text{in lb}}{\text{sec hp}}}\right) = 38 \text{ hp}$$

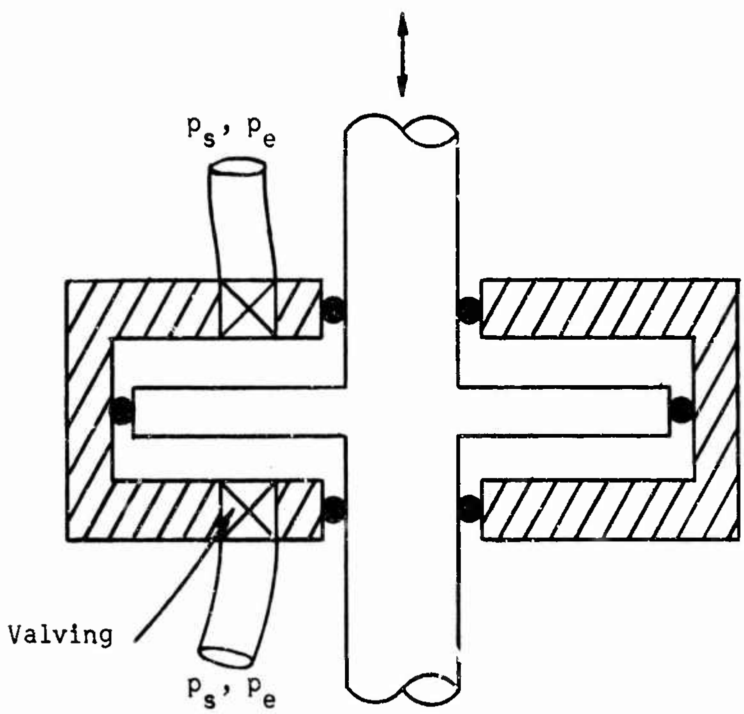


Figure B-1. Fluid Actuator Controlled by External Valving

If p_s is greatly reduced, the piston area and the supply line area will have to be increased so that the increase in v will partially offset the decrease in p_s, and $\mathbb{P}_c$ will still be large. Moreover, the increased inertial forces required to accelerate a large piston will reduce the output of the device.

In view of the above considerations, pneumatic piston-type actuators were concluded to be unacceptably inefficient, and they were not considered further.

APPENDIX C

HYDRAULIC PISTON ACTUATOR USING EXTERNALLY-DRIVEN VALVING

This appendix investigates the feasibility of using a suitably valved hydraulic, piston-type actuator to excite the drill rod. The valving is externally driven at the resonant frequency of the drill rod. Due to the high bulk modulus of the oil, fluid compressibility power losses are acceptably small, i.e., about 20%. Consequently, high fluid pressures can be used with resulting compact, low-inertia actuators. Whereas the required power level is well within the state-of-the-art of hydraulics, specialized valving and careful design are required to achieve the high-frequency response that is required.

The simplest way to port liquid into and out of the two piston chambers is to use a rotating valve that is enclosed in a housing containing the various flow passages. The geometry is arranged so that equally spaced holes in the valve periphery alternately connect the supply and exhaust ports to each piston chamber in the proper sequence.

There are a number of ways to implement the rotary valve. Figure C-1 shows one simple configuration which utilizes the following parameters:

p_s = 2000 psi

N = 10,000 rpm

d = 0.1 in

r = 0.5 in

D = 2 in

ℓ = 0.7 in

n = 8 holes in the disk

v = total compressed volume = 0.4 in^3

x_m = 0.0015 in = piston displacement amplitude

ρ = oil density = $0.85 \times 10^{-4} \dfrac{\text{lb sec}^2}{\text{in}^4}$

A_p = 3 in^2

h = 0.001 in

A_ℓ = 0.2 in^2

ρ_p = piston density = 0.1 lb/in^3

p_e = 15 psia

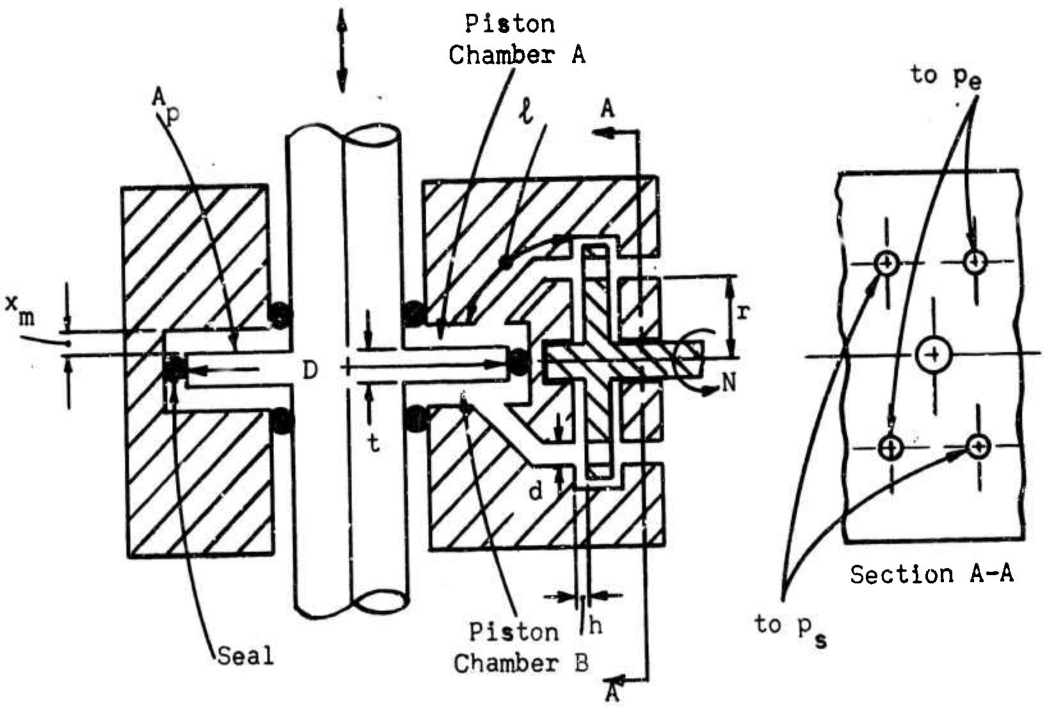


Figure C-1. Hydraulic Actuator

The resonant frequency Ω_ℓ of the transmission line of length ℓ is

$$\Omega_\ell = \frac{c}{4\ell} = \frac{5 \times 10^4 \ \frac{in}{sec}}{(4)\ (0.7\ in)} = 18{,}000 \text{ cps} \qquad \text{(C-1)}$$

where $\quad c \quad = \quad$ velocity of sound in oil

From Reference (7), the surge frequency of this line when it is terminated by the actuator volume is about 13 kcps. Since this is an order of magnitude larger than the excitation frequency, a lumped parameter analysis can be used to predict the system dynamics. The time constant, τ_1 of the valve orifice and trapped volume capacitance with the piston blocked, is approximately $1/2 \times 10^{-4}$ seconds. This implies a cutoff frequency that is 2 1/2 times larger than the exciter frequency. With the piston blocked, the natural frequency, ω_1, of the line inertance and the trapped volume capacitance is 8000 cps. Thus, the dynamic response of the rotary disk valve hydraulic system is adequate.

If the actuator thickness, t, is assumed to be 1/2-inch, then the inertia force required to accelerate the piston, F_i, is

$$F_i = \left(\frac{\rho_p At}{g}\right) (2\pi f_N)^2 x_m \qquad \text{(C-2)}$$

$$= \left(\frac{0.1\ lb}{in^3}\right) \frac{(3\ in^2)\ (1/2\ in)}{\left(386\ \frac{in}{sec^2}\right)} \left(72 \times 10^6\ \frac{rad^2}{sec^2}\right)(1.5 \times 10^{-3}\ in)$$

$$= 42 \text{ lb}$$

The advantages of using high working pressures are apparent from this calculation which shows that F_i is less than 1% of the total piston force.

The flow, Q_T, that must be supplied is composed of three components:

(a) The actuator flow, Q_a, which is equal to the volume swept by the actuator times the excitation frequency.

$$Q_a = 2\ f_N\ A\ 2\ x_m = 24\ in^3/sec \qquad \text{(C-3)}$$

(b) The compressibility flow, Q_c, which is proportional to the product of the total compressed volume, the pressure fluctuation and the frequency, divided by the bulk modulus, of the fluid.

$$Q_c = f_N v(p_s/B) = 5\ 1/2\ in^3/sec \qquad (C\text{-}4)$$

(c) The flow, Q_ℓ, which leaks past the nozzles in the spinning disk.

A more involved calculation accounting for orifice flow and laminar flow pressure drops indicates that

$$Q_L = 2\ 1/2\ in^3/sec \qquad (C\text{-}5)$$

Thus, 75% of the total flow that is supplied does useful work. If p_s is lowered by a factor of two, this percentage can be increased since Q_c and Q_L will then be a smaller fraction of Q_T.

The fluid inertia of the transmission line will result in a line pressure drop, Δp_i, which can be estimated as

$$\Delta p_i = \left(\frac{\rho \ell}{A_\ell}\right) \frac{d Q_a}{dt} \qquad (C\text{-}6)$$

where $$\frac{d Q_a}{dt} \cong (2\pi f_N)^2 (A_p x_m) \qquad (C\text{-}7)$$

$$= (72 \times 10^6 \frac{rad^2}{sec^2})(3\ in^2)(1.5 \times 10^{-3}\ in)$$

$$= 3.2 \times 10^5\ in^3/sec^2$$

The pressure drop is thus

$$\Delta p_i = (0.85 \times 10^{-4} \frac{lb\ sec^2}{in^4}) \frac{(3/4\ in)}{(.2\ in^2)} (3.2 \times 10^5 \frac{in^3}{sec^2})$$

$$\cong 100\ psi$$

This calculation shows that the line length must not be too long and its area must not be too small or else Δp_i can become an appreciable fraction of p_s.

Another factor that must be considered in the actuator design is squeeze film damping. It is anticipated that this effect can be reduced to an acceptable level either by cutting grooves in the piston, or by increasing the axial clearance between the piston and the ends of the chamber.

The disk may be driven either directly by the gasoline engine that drives the pump, or by a hydraulic turbine. The latter technique would enable the operator to easily and accurately adjust the disk speed by controlling the turbine flow.

The rotating valve can be axially supported in the housing by hydraulic thrust bearings that are supplied by leakage flow from the supply nozzles. The axial clearance, h, of 0.001 inches (see Figure C-1) will restrict this leakage flow to an acceptable amount. The disk is radially supported either by ball bearings or by hydraulic thrust pads located around the housing periphery.

The primary design considerations for the reservoir are small size, light weight, and adequate heat dissipation. A 3 gallon reservoir has a volume of approximately 1/2 ft^3. If this is obtained with a rectangular tank with dimensions 1 ft x 1 ft x 1/2-ft, the heat radiating area, A, is approximately 4 ft^2. A free convection heat transfer coefficient, h, of 1 BTU/hr ft^2 ^oF, and a heat generation rate, q, of 1 horsepower (2500 BTU/hr) result in a reservoir surface temperature, T_s, of

$$T_s = T_a + \frac{q/A}{h} \tag{C-8}$$

$$= 32^o + \frac{2500 \text{ BTU/hr}}{4 \text{ ft}^2} \; \frac{1}{1 \dfrac{\text{BTU}}{\text{hr ft}^2 \, {}^o\text{F}}} \cong 650^o\text{F}$$

where T_a = ambient temperature 32^oF

If the duty factor of the device is 1/4, the surface temperature will be 160^oF which is acceptable. If the duty cycle is larger, additional heat transfer area or forced cooling will likely be required.

It will take about 2/3 hp for cold startup at - 40^oF in order to turn the disk valve at 10,000 rpm while it is immersed in oil. However, this power is reduced to less than 1/50 hp once the oil warms up to 100^oF.

On the basis of the preceding considerations, it was concluded that a hydraulic piston-type actuator utilizing a rotary valve appears to be an extremely promising way to excite the drill rod. Consequently, this system was considered further with regard to the auxiliary equipment that it requires.

APPENDIX D

HYDRAULIC GEAR PUMP PRESSURE PULSE GENERATOR

This appendix examines the feasibility of exciting the drill rod with pressure pulses that are generated by a specially designed gear pump. The pressure pulses are generated when the meshing gear teeth trap a volume of fluid and compress it. The high-pressure trapped volume communicates with actuator volume through axial ports that are machined into the pump. A schematic arrangement of the pump and the ports is shown in Figure D-1.

Conventional gear pumps do not experience large pressure fluctuations because they are provided with passages that permit the trapped fluid to escape from the gear meshing area to the discharge area. In the presently envisioned adaptation of a gear pump, these escape passages are blocked off and ports A, A' connected to the two actuator volumes are substituted. These ports supply high-pressure pulses to the actuator volumes. An additional pair of ports labeled B, B' enables each actuator volume to discharge to low pressure when the opposite volume is undergoing a pressure buildup. The ports are positioned so that each actuator volume is only connected to one port at a time.

The device is sized by evaluating the following equation of continuity of mass, assuming sinusoidal conditions:

$$\frac{d\,V_G}{dt} = \frac{V}{B}\,\frac{dP}{dt} + A\,\frac{dx}{dt} + Q_L \tag{D-1}$$

where $\frac{d\,V_G}{dt}$ = rate of change of trapped gear volume = $(\Delta V)2f_N$

ΔV = volume displaced by two meshing gear teeth

B = fluid bulk modulus

A = actuator area

x = piston displacement

Q_L = leakage flow

V = pressurized volume of actuator and connecting lines and gear meshing area

f_N = operating frequency

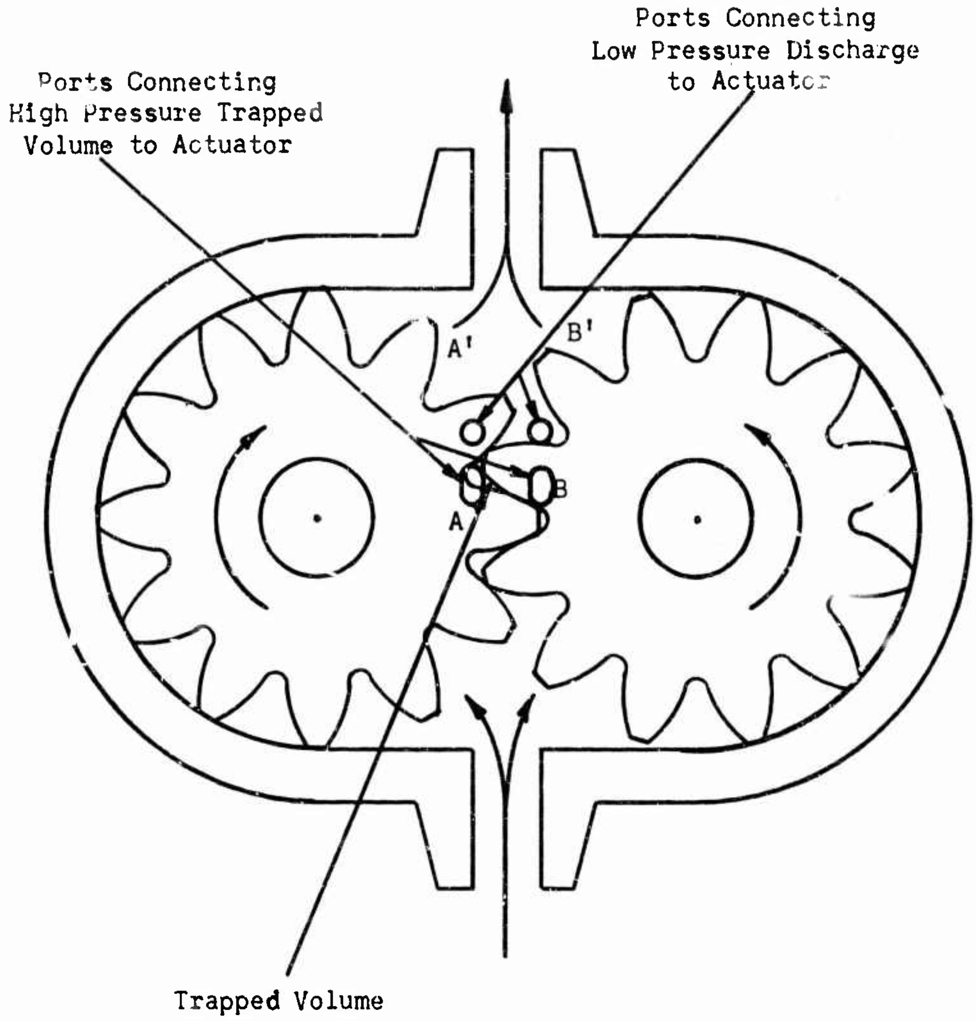


Figure D-1. Hydraulic Gear Pump Pressure Pulse Generator

The following typical parameter values are assumed:

$$B = 200{,}000 \text{ psi}$$

$$V = 0.1 \text{ in}^3$$

$$x_{max} = 0.003 \text{ in}$$

$$p_{max} = 1000 \text{ psi}$$

$$f_N = 1350 \text{ cps}$$

$$Q_L = 1 \text{ gpm}$$

Substitution of these values into Equation (D-1) yields

$$\Delta V = 0.0125 \text{ in}^3$$

If the gear is 1 inch thick, the required meshing area is approximately 0.1 inches square — a reasonably sized gear. The device could use two ten-tooth gears rotating at 9000 rpm.

The modified pump has a small nominal load at its discharge. Frequency "tuning" of the rotation speed of the gears can be obtained by adjusting this load.

There are a number of practical problems that exist with the gear pump pulse generator. It will be difficult to control the amplitude and shape of the pressure pulses since these will be strongly influenced by the local geometry of the mating gear teeth. High bearing loads, and excessive leakage flows may also prove troublesome. Some form of pressure relief will likely be required to prevent damage to the device in case the bulk modulus of the fluid is larger than expected, leakage flows are smaller than expected, or if the piston becomes blocked.

The hydraulic gear pump pressure pulse generator combined with a piston-type actuator appears to be a feasible way to excite the drill rod. However, a substantial amount of development work which is beyond the scope of the present contract would be necessary in order to produce a workable device. Consequently no further consideration was given to this technique for purposes of constructing a laboratory model exciter.

APPENDIX E

UNSTABLE FLUID BEARING ACTUATOR

This appendix evaluates the feasibility of using an unstable air bearing to excite the drill rod.[8] A typical arrangement is shown schematically in Figure E-1. For small displacements, the relation between the bearing force, F, and its displacement, x, is of the form

$$F = \frac{k\,(\tau_1\,S + 1)x}{(\tau_2\,S + 1)} \qquad \text{(E-1)}$$

where

- k = the static bearing stiffness
- τ_1 = a lead time constant arising from squeezing action between the piston and the bearing
- τ_2 = a lag time constant
- S = the Laplace transform operator

If τ_1 is neglected, the power that the fluid puts into the drill rod is given by[1]

$$\mathbb{P}_b \cong \frac{1}{4}\,\frac{F_m}{h}\,(2\pi f_N)\,x_m^{\,2} \qquad \text{(E-2)}$$

where

- F_m = maximum force (= 5000 lb)
- h = nominal bearing clearance
- x_m = piston travel (= 0.003 in)
- f_N = drill rod natural frequency (= 1350 cps)

From Equation (E-2), in order for $\mathbb{P}_b$ to be 5 horsepower, the required clearance is

$$h = 0.003 \text{ inches}$$

If a supply pressure of 1500 psi is chosen, a piston diameter, D, of 3 inches is required to obtain F_{max}. The air flows out of the system through the two curtain areas, A_c.

$$A_c = \pi\,D\,h \qquad \text{(E-3)}$$

$$= (\pi)\,(3)\,(0.003) = 0.028 \text{ in}^2$$

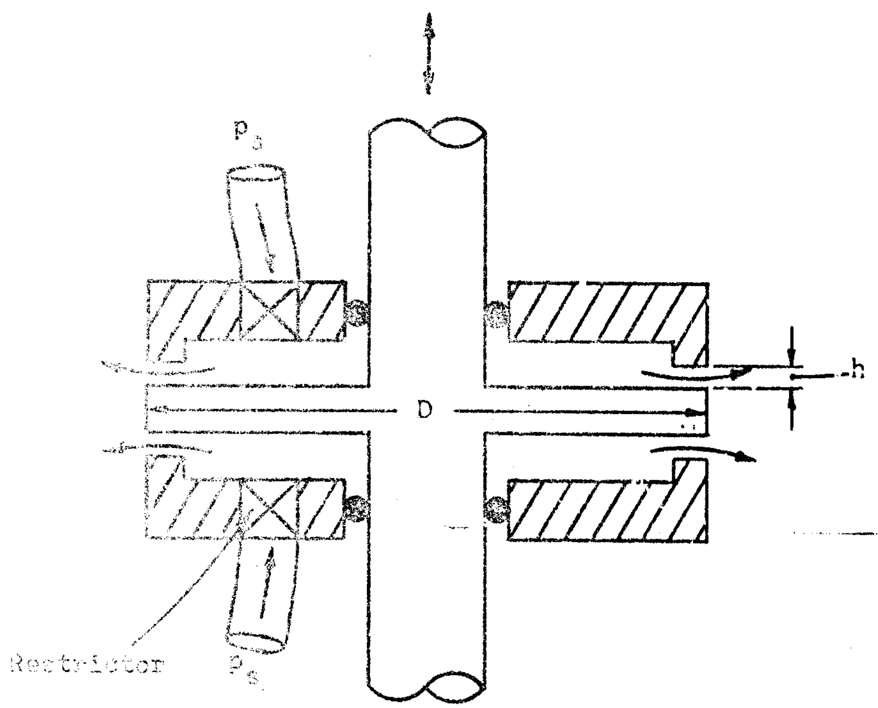


Figure E-1. Unstable Fluid Bearing

Following usual practice, the quiescent chamber pressures are made equal to $p_s/2$, which results in a flow rate, w, and a power consumption, $\mathbb{P}_c$, of

$$w = 0.95 \text{ lb/sec}$$

$$\mathbb{P}_c = 170 \text{ hp}$$

Quite clearly, this is an unacceptably inefficient way to excite the drill rod. Since decreasing p_s by an order of magnitude still results in large losses, this type of exciter was not considered further due to its inefficiency.

If hydraulic oil is used in the configuration shown in Figure E-1, the analysis is identical to the pneumatic case except that the flow, w, and the power consumption, $\mathbb{P}_c$, are evaluated differently. If each curtain area is

$$A_c = 0.028 \text{ in}^2$$

the total volume flow, Q, is given by

$$Q = 2\ c_d A_c \sqrt{\frac{2}{\rho}\left(\frac{p_s}{2}\right)} \tag{E-4}$$

$$= (0.9)(0.056)\left(\frac{1500 \text{ lb}}{0.85 \times 10^{-4} \text{ in}^2 \frac{\text{lb sec}^2}{\text{in}^4}}\right)^{1/2}$$

$$= 205 \text{ in}^3/\text{sec}$$

where c_d = discharge coefficient

The power consumption, $\mathbb{P}_c$ is

$$\mathbb{P}_c = p_s Q \tag{E-5}$$

$$= \left(1500 \frac{\text{lb}}{\text{in}^2}\right)\left(205 \frac{\text{in}^3}{\text{sec}}\right)\left(\frac{1}{6600 \frac{\text{in lb}}{\text{sec hp}}}\right)$$

$$= 46 \text{ hp}$$

This is an unacceptably large power consumption. For fixed F_m, x_m, and piston quiescent pressure ratio (0.5 is the optimum value), the flow Q is fixed and the efficiency of the device, η, is of the form

$$\eta = \frac{(2\pi f_N)^2}{4} K_1 \left[\frac{\frac{p_s}{2\pi f_N} - K_1}{p_s^2} \right] \qquad \text{(E-6)}$$

where $$K_1 = \frac{F_m x_m}{Q} = 0.075 \frac{\text{lb sec}}{\text{in}^2}$$

This expression takes into account the previously neglected parasitic effects of the time constant τ_1. This efficiency is unacceptably low for all values of p_s in the range of interest.

The unstable bearing type of exciter has the fundamental disadvantage of requiring a through flow which is greatly in excess of the useful flow that powers the piston. Due to its resulting inefficient operation, this type of system was eliminated from further consideration.

APPENDIX F

HYDRAULIC ACTUATOR DRIVEN BY A FLUIDIC OSCILLATOR

Fluidic components could be used to generate a self-excited oscillation that is amplified, and used to power a piston-type actuator. Alternatively, a high-power oscillator might be used to directly drive the piston. The anticipated efficiency of this type of device is relatively low due to large leakage flows, and low pressure recovery in the receiver ports. Maintaining the oscillator frequency equal to the drill rod resonant frequency would probably be difficult. It is questionable whether the switching speed of a 5 hp jet is rapid enough to achieve a 1300 cps oscillation frequency. Moreover, a substantial amount of development work would be necessary to produce a working device, since the desired power level is greatly in excess of those of present fluidic systems. Consequently, this type of system was dropped in favor of the more straightforward externally-driven rotary valve exciter.

APPENDIX G

TWO-PHASE FLUID TRANSDUCER

A two-phase fluid making use of rapid vapor bubble formation and collapse could conceivably be used in a piston-type of actuator. A source of heat is required to generate the vapor bubbles, and a heat sink is required to remove heat to collapse the bubbles. A spark discharge or some other fast acting heating element is necessary for the heat source.

An upper bound on the efficiency of this type of exciter was determined by analyzing a single cycle of operation and neglecting all dynamic effects. It was conservatively assumed that all of the heat went into vaporization of a gas bubble. The resulting loss of liquid volume raises the liquid pressure. The pressure is assumed to be uniform throughout the fluid, and to remain constant while the fluid moves the piston. The efficiency, η, which is taken to be the quotient of the work performed on the piston divided by the heat input, is given by

$$\eta \cong \frac{\bar{R}\, T_g}{M\, h_{fg}\, (778)} \qquad \text{(G-1)}$$

where

$\bar{R}$ = gas constant = 1545 ft lb/^{o}R lb mole

M = molecular weight of the fluid

h_{fg} = enthalpy of vaporization in BTU

T_g = temperature of gas bubble ^{o}R

The idealized analysis is not valid for temperatures approaching the critical point of the fluid since the perfect gas law no longer applies. It should be noted that h_{fg} is a function of temperature which decreases to zero as the critical point is approached.

If water is used and the system is operated near a T_g of 700^{o}R, then Equation (G-1) yields

$$\eta = \frac{\left(\frac{1545\ \text{ft lb}}{^{o}\text{R lb mole}}\right)(700^{o}\text{R})}{\left(970\ \frac{\text{BTU}}{\text{lb}}\right)(18\ \text{lb/lb mole})\left(778\ \frac{\text{ft lb}}{\text{BTU}}\right)} \qquad \text{(G-2)}$$

$$\cong 8\%$$

Increasing T_g to $1000^{\circ}R$ does not significantly alter η. The upper bound on the efficiency is not greatly increased by using more volatile fluids, since they usually have large molecular weights and they decompose at high temperatures.

The two-phase fluid transducer has numerous practical difficulties. Due to its low efficiency, the large amount of heat that must be removed will require an elaborate cooling system involving forced convection liquid cooling combined with a large radiator. Nonequilibrium conditions in the actuator will result in nonuniform pressures that reduce the efficiency of the device. If a spark discharge is used, fluid degeneration may result and it will be difficult to control the process since changing fluid properties will change the spark discharge characteristics. The high voltage necessary to generate the spark will be a safety hazard, and will require an additional auxiliary system.

Since the two-phase fluid transducer did not appear to offer any particular advantages over other transducers, it was not considered further due to its low efficiency.

APPENDIX H

PIEZOELECTRIC EXCITER

This appendix examines the feasibility of using a piezoelectric transducer to excite the drill rod. Piezoelectric forces result from the tendency of certain materials to strain in the presence of an electric field. The most commonly used piezoelectric material for high-power applications is PZT-4, a ceramic composite that can be fabricated into various standard shapes.[9]

A common type of transducer configuration that is used in sonar and ultrasonic cleaning applications [2, 3] is shown schematically in Figure H-1. It consists of a piezoelectric transducer element sandwiched between two masses which are approximately one quarter wavelength long. The entire system is resonated in its free-free mode at 1350 cps. A typical maximum dynamic stress for PZT-4 ceramic is

$$\sigma_{max} = 3500 \text{ psi}$$

Taking this stress into account, the maximum transducer displacement is given by

$$\delta = \frac{L\,\sigma_{max}}{E_Y} = 3.5 \times 10^{-4}\,L \qquad \text{(H-1)}$$

where E_Y = Young's modulus for PZT-4 = 10^7 psi

L = total active length for the exciter

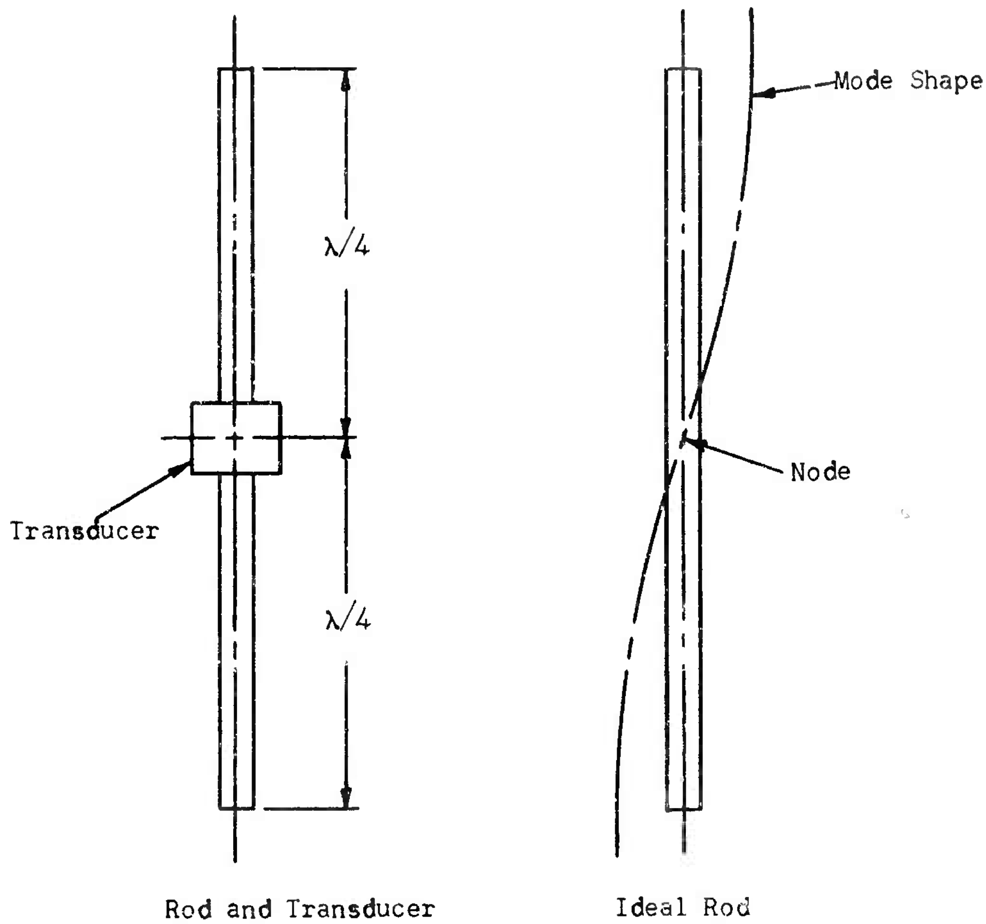
The transducer efficiency is on the order of 85%, and a typical value of power per unit volume is

$$P_d = 150 \text{ watts/in}^3\text{/KC}$$

$$= 200 \text{ watts/in}^3 \text{ at } 1350 \text{ cps}$$

If 5 hp or 3750 watts is required for the output power, then for this value of P_d the total volume of piezoelectric must be

$$V_c = \frac{3750}{200} = 18.75 \text{ in}^3$$



λ = wavelength at excitation frequency

Figure H-1. Piezoelectric Sandwich Transducer

For a typical piezoelectric wafer size having a 2 inch O.D. and a 1.25-inch I.D., the required active length to provide this volume becomes

$$L \cong \frac{(18.75)\ (4)}{\pi\ (4 - 1.56)} = 10 \text{ inches}$$

The peak force for the typical maximum dynamic stress, σ_{max}, of 3500 psi is

$$F_e = \sigma_{max}\ A = \frac{3500\ \pi\ (4 - 1.56)}{4} = 6700 \text{ lbs} \qquad \text{(H-2)}$$

and the maximum displacement from Equation (H-1) is

$$\delta = 3.5 \times 10^{-3} \text{ inches}$$

The electric field required to produce 3500 psi in PZT-4 is approximately

$$E_f = 4 \text{ k volts/cm rms}$$

This electric field can be achieved by constructing the transducer from a stack of 100 0.1 inch thick wafers that are excited electrically in parallel by 1000 volts.

Piezoelectric ceramics are brittle materials which are easily fractured by tensile stresses. Figure H-2 shows the application of a compressive bias stress to these elements to prevent tensile stresses. Belleville washer springs are used to produce the bias force since their low effective spring stiffness will not significantly affect the transducer resonant frequency.

The maximum allowable compressive bias stress of 3000 psi for PZT-4 imposes a limit on the maximum allowable dynamic strain in the ceramic. The actual dynamic strain depends on the load that is presented to the transducer. The load is influenced by the permafrost conditions and by the force with which the operator pushes on the drill rod. Since these factors may vary considerably in operation, the transducer will not always be properly matched to its load. Thus, unless preventative measures are taken, the oscillation amplitude may exceed the allowable limit, and fracture the ceramic. This can be avoided if the excitation voltage or current is controlled. One technique utilizes an accelerometer that is mounted on the transducer. The accelerometer output, which is proportional to displacement, can be used in a feedback circuit to control the excitation voltage. Another technique involves limiting the current going into the transducer since the transducer strain is proportional to the current.

Electrical energy that is put into the transducer is converted into useful mechanical power and parasitic heat losses. The losses due to dielectric heating and internal mechanical losses in the ceramic are on the

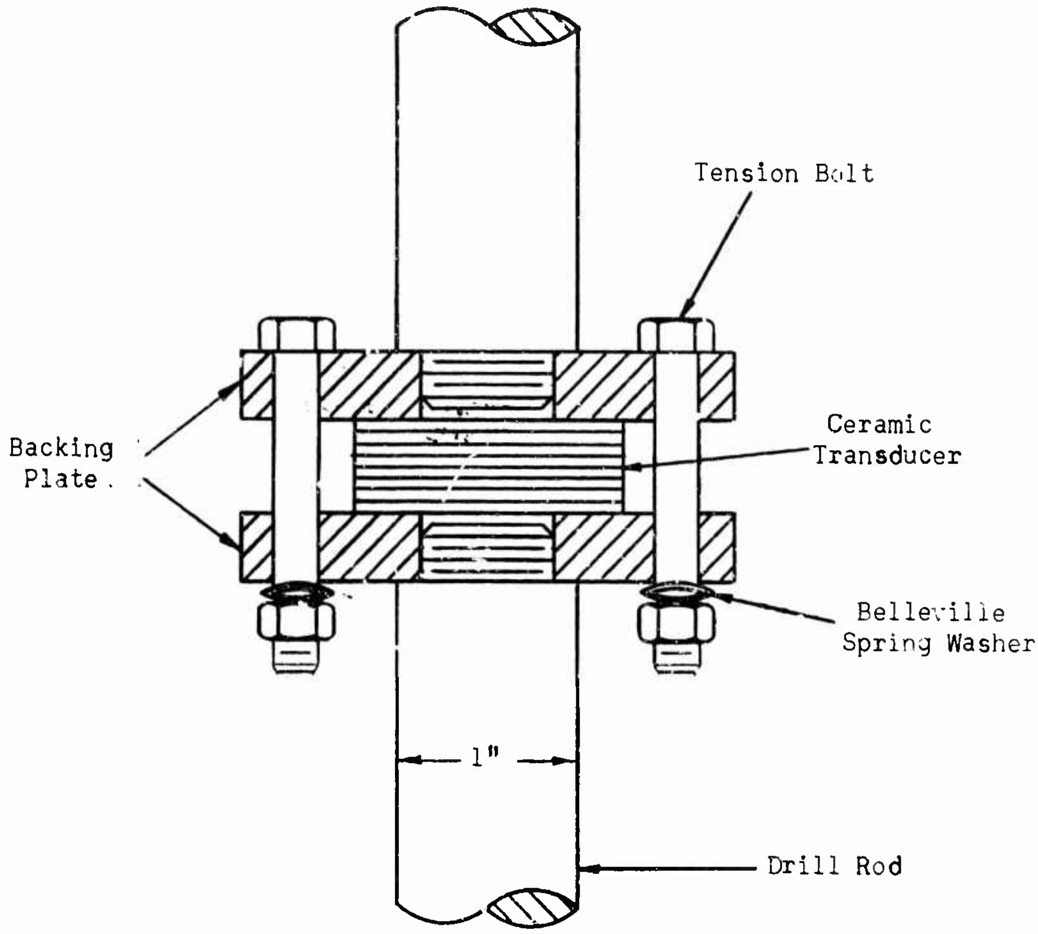


Figure H-2. Piezoelectric Transducer with Bias Stress

order of 5% of the total input power. External mechanical losses in the transducer mounting structure typically amount to 10%.

When 4 KW of electrical power is applied to an 85% efficient transducer, 0.6 KW (2000 BTU/hr) of heat must be removed. The previously described transducer has a peripheral heat transfer area of approximately 1/2 ft^2. For a working surface temperature, T_s, of 200°C, and an ambient temperature, T_a of 0°C, the required heat transfer coefficient, h, is

$$h = \left(\frac{g}{A}\right) \frac{1}{(T_s - T_a)} = \frac{(2000 \frac{BTU}{hr})}{(\frac{1}{2} ft^2)} \left(\frac{1}{360^\circ F}\right) \tag{H-3}$$

$$= 11 \frac{BTU}{hr\ ft^2\ {}^\circ F}$$

Forced gas convection will be necessary to achieve this heat transfer coefficient. Thus, a fan will be required.

It can be shown that the highest temperature in the interior of the ceramic will be 280°C which is within the allowable operating range of PZT-4 (3).

On the basis of the preceding considerations, it was concluded that a piezoelectric exciter is a promising way to excite the drill rod.

APPENDIX I

MAGNETOSTRICTIVE EXCITERS

Exciting the drill rod with forces induced by the magnetostrictive effect is investigated in this appendix. These forces result from the tendency of certain materials (nickel is commonly used) to strain in the presence of a magnetic field. Several companies in the United States and Great Britain use magnetostrictive transducers for ultrasonic drilling work.

Figure I-1 shows a typical configuration for a magnetostrictive transducer that is composed of a stack of nickel laminations (10). The stack is usually designed to resonate in its half wavelength mode. A half wavelength resonating tapered horn that magnifies the transducer displacement amplitude is placed between the stack and the drill bit.

The following figures of merit are based on experience with existing magnetostrictive devices (11):

$$\text{Efficiency, } \eta = \frac{\text{mechanical output power}}{\text{electrical input power}} \cong 35\% \qquad \text{(I-1)}$$

$$\text{Power Density, } P_w = \frac{\text{mechanical output power}}{\text{transducer weight}}$$

$$\cong \frac{60 \text{ watts}}{\text{lb}} \qquad \text{(I-2)}$$

The requirement for a 5 hp output is a factor of three more than the capability of present devices. If the above numbers are assumed to apply to a 5 hp transducer, then the device would require 15 horsepower (11 kilowatts) of electrical input and would weigh over 60 lb. The low power conversion efficiency would necessitate a forced convection liquid cooling system to dissipate 10 horsepower, i.e., 7 BTU/sec. The cooling system with its pump, fluid, and radiators will significantly increase the exciter weight.

A preliminary design for a magnetostrictive transducer was prepared which resulted in a unit that is approximately 8 x 8 x 12 inches, has an excitation frequency of 6 KC, and weighs 90 lbs. A 20-inch long tapered horn increases the displacement amplitude from 0.0005 inches to 0.0027 inches.

When they are compared with piezoelectric devices, advantages that are generally cited for magnetostrictive devices are that they are more rugged, more easily cooled and their input impedance is lower, making them more easily driven with solid-state components. However, while magnetostrictive materials are not prone to the type of brittle fracture that can occur in

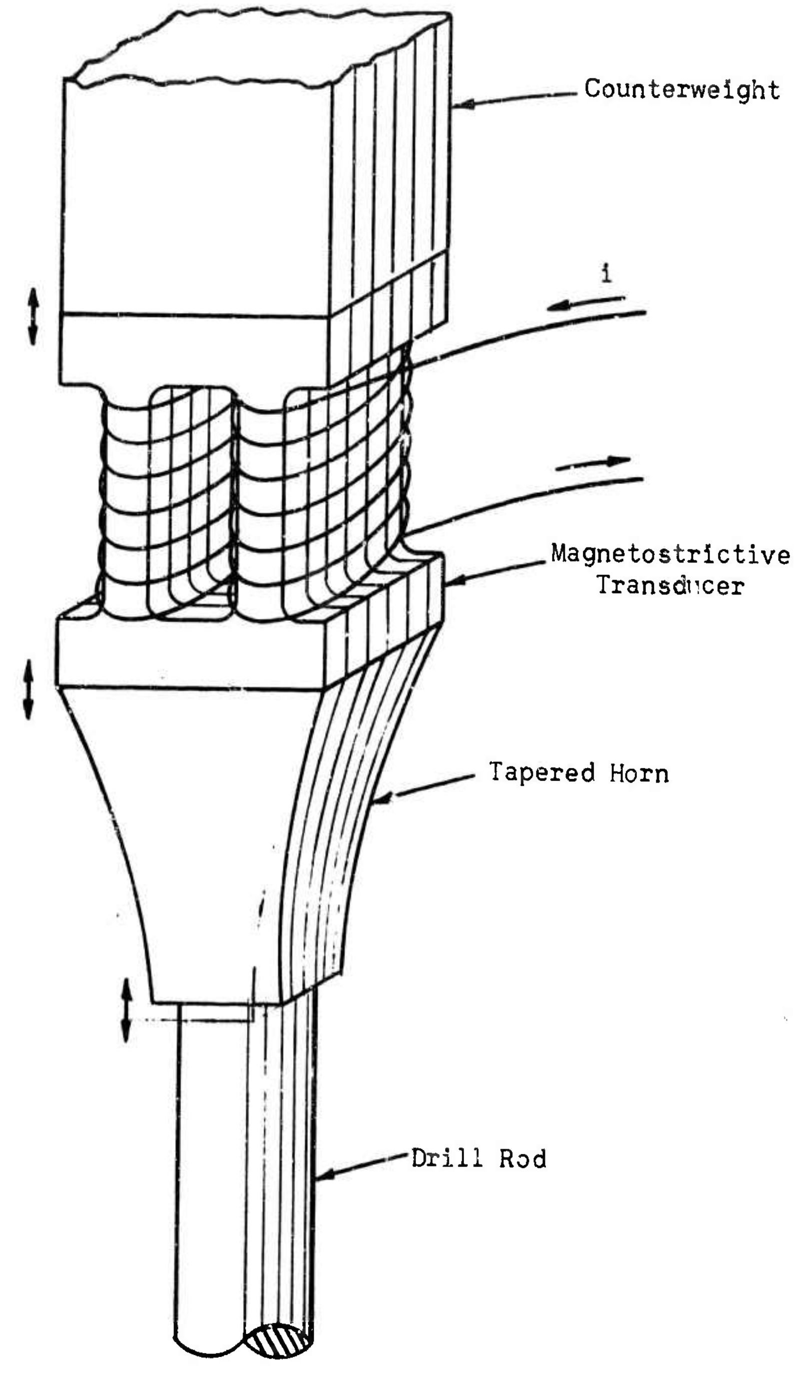


Figure I-1. Magnetostrictive Transducer

piezoelectric ceramic materials, nickel-steel has a low fatigue strength (10,000 psi) which mitigates this advantage. Precompression techniques have been developed for piezoelectric transducers which gives them comparable durability. Easier cooling of nickel-steel due to its relatively high conductivity is offset by the fact that it generates four times as much heat as a ceramic material. In general, in the high-power ultrasonics field there has been a shift in emphasis away from magnetostrictive towards piezoelectric devices. However, if more efficient magnetostrictive materials are developed, this trend might well be reversed.

Although a magnetostrictive exciter appears feasible, the low efficiency of present devices, combined with a high predicted weight, make this type of transducer less desirable than other electromechanical systems, e.g., the piezoelectric system. Consequently, magnetostriction was not considered further.

APPENDIX J

ELECTROMAGNETIC EXCITERS

The feasibility of using electromagnetic forces to excite the drill rod is examined in this appendix. The basic problem uncovered in considering the various electromagnetic transducers that might be used (e.g., torque-motors, solenoids, linear electric motors) is the fundamental limitation of less than 100 psi in the equivalent pressure which they can produce due to magnetic saturation. For practical designs, the attainable force per unit is usually much lower than this limit. Consequently, large areas are needed to generate the required force levels. As a result, most of the magnetic force is used up overcoming the large inertial forces of the massive moving part. Calculations are presented for several typical configurations which illustrate this point.

A flat-faced armature tractive magnet was investigated to see if it could produce an external force, F_e, of 5000 lb and a 0.003 inch deflection at 1300 cps. The magnet force, F_m is given by

$$F_m = \left(\frac{B^2}{72}\right) A \tag{J-1}$$

where
- A = armature area, in^2
- B = magnetic flux density in kilomaxwells/in^2
- F_m = magnetic force - lb

A typical value of B is 85 kmax/in^2 which results in an effective magnetic pressure, p_m, of 100 psi. If it is assumed that an inertia force, F_i, of 5000 lb is required to accelerate the armature, then

$$F_m = F_e + F_i = 10{,}000 \text{ lb} \tag{J-2}$$

and the armature area is

$$A = \frac{F_m}{p_m} = 100 \text{ in}^2 \tag{J-3}$$

In order to keep the armature deflection less than 0.002 inches, its thickness must not be less than 1 inch, resulting in an armature weight of 28 lb. The armature displacement amplitude, x, can be solved from the relationship

$$F_i = \left(\frac{W}{g}\right) (2\pi f_N)^2 x \tag{J-4}$$

where
- f_N = the drill rod natural frequency (1350 cps)

The resulting amplitude is less than 0.001-inches which is unacceptably small. This situation is not significantly improved by altering the armature thickness within a reasonable range. Consequently, it was concluded that this type of magnet is not suitable for the present application.

Commercially available vibratory shake tables use a moving-coil type of magnet. Units in the 5 hp range are far too heavy for the present application. A typical device (manufactured by the MB Electronics Company) capable of producing 5000 lbs of force weighs 3000 lbs. Thus, as a result of their prohibitive weight, moving-coil type magnets have been eliminated from further consideration.

A leakage-flux solenoid type of magnet was also analyzed. Using relationships that are derived in Reference (12), an excessively large moving volume of iron is required, which reduces the amplitude to less than 0.0002 inches which is unacceptable.

Linear induction motors were investigated and found to have unacceptably large current requirements. According to Reference (13), current densities on the order of 2×10^5 amp/in^2 are necessary to produce a 100g linear acceleration. The present application has a 750g sinusoidal acceleration which would require even larger current densities.

As a result of these studies, it was concluded that electromagnetic devices are either incapable of producing the required force and stroke at the high operating frequency, or they are unacceptably large and heavy for this application. Consequently they were not considered further.

APPENDIX K

ELECTROSTATIC TRANSDUCERS

This appendix investigates the feasibility of utilizing electrostatic forces between charged surfaces to excite the drill rod. However, an electrostatic transducer that produces the required force and stroke would be impractical since the charged surfaces would be extremely large. This can be seen by considering the force, F_e, existing between two flat capacitor plates of area A which are separated by an air gap of length x. If a voltage V exists between the two plates, it can be shown that F_e is given by

$$F_e = -\frac{\varepsilon V^2 A}{2x^2} \qquad \text{(K-1)}$$

where $\varepsilon = 8.85 \times 10^{-12}$ farad/meter, the dielectric constant of air

If the following values are assumed,

$$V = 10^4 \text{ volts}$$

$$x = 0.004 \text{ in}$$

$$F_e = 1800 \text{ lb}$$

then from Equation (K-1), the required surface area is

$$A = 3 \times 10^8 \text{ in}^2 \qquad \text{(K-2)}$$

If ε is increased by using a fluid between the plates, A is still excessive, and large power losses will result from squeeze damping action.

Due to their prohibitive size, electrostatic actuators were not considered further.

APPENDIX L

AUXILIARY EQUIPMENT REQUIRED FOR THE HYDRAULIC EXCITER DRILLING SYSTEM

This appendix investigates the auxiliary equipment requirements of the hydraulic drilling system. This equipment consists of the following components:

- pump
- prime mover to drive the pump
- reservoir
- hydraulic oil
- pressure relief valve
- connecting hydraulic lines

For portable operation in an arctic environment, a gasoline engine is the cheapest, most straightforward prime mover. Table L-I indicates the weights and costs of several gasoline engines and hydraulic pumps.

The fluid weight of a hydraulic system using 3 gallons of oil will be 20 lb, and a 3 gallon reservoir fabricated from 1/8-inch thick aluminum plate will weigh 7 lbs. The valve housing, piston housing, and mounting structure will weigh approximately 10 lb while the piston, disk valve, and disk speed control should not weigh more than 4 lbs.

Table L-I

Hydraulic Components

A - Gear Pumps

Flow Rate (gpm)	at	Maximum Speed (rpm)	Maximum Pressure (psi)	Weight (lb)	Manufacturer	Cost
4		2,500	2,000	3	Webster	$ 30
4		7,000	1,000	2-1/2	Pesco	400

B - Gasoline Engines

Maximum Power (hp)	Speed (rpm)	Weight (lb)	Manufacturer	Cost
10	10,000	11	McCulloch	$140
6	3,600	36	Tecumseh Lauson	90
6	3,600	43	Briggs and Stratton	100

APPENDIX M

AUXILIARY EQUIPMENT REQUIRED FOR

THE PIEZOELECTRIC DRILLING SYSTEM

This appendix investigates the auxiliary equipment requirements for the piezoelectric drilling system. The piezoelectric system requires 1000-volt, 4-ampere, 1350-cps electrical excitation. This can be provided either by a solid-state driver or by a high-frequency rotary generator. A solid-state driver requires a supply voltage that could be obtained from a dc, 60 cps or 400 cps generator that is driven by a gasoline engine. Table M-I shows the weights and costs of representative engines and generators that are commercially available.

If a solid-state electric driver is used, it will have to be specially developed since a reasonably sized 4-Kv-a unit capable of delivering 1000 volts at 1600 cps is not an off-the-shelf item. A supplier of this type of component estimates that a specially designed driver utilizing silicon controlled rectifiers that are supplied with 200 volts dc will have the following parameters:

cost	–	$500
weight	–	50 lbs
volume	–	1/2 ft^3
efficiency	–	90%

The oscillator would likely be contained, along with the prime power source (i.e., a motor-driven generator), in a separate portable power pack. This power pack would be connected to a transformer, mounted on the drill rod, by means of a cable. It would be undesirable to package the oscillator on the drill rod where it would be subjected to the intense vibration environment.

The functions of the electrical [illegible]ator and the solid-state, high-power oscillator can be combined in a high-frequency rotary generator. Unfortunately, there are very few off-the-shelf units that generate 4kva in the 1000 cps to 1600 cps frequency range. There are a number of basic problems associated with directly driving the transducer with a high-frequency generator. Since the generator frequency varies directly with the generator speed, the gasoline engine will require an accurate, high-resolution speed control system to maintain the drill rod in resonance To prevent a failure in an unloaded transducer, it will probably be necessary to switch off the excitation voltage when the transducer strain amplitude reaches a critical value. The resulting large load disturbances will increase the difficulty of maintaining control over engine speed. In addition, it will be necessary to have a gear train between the gasoline

Table M-I

Piezoelectric Power Source Components

A - Gasoline Engine-Generator Sets

Voltage	Power (Kw)	Weight (lb)	Manufacturer	Cost
250 v dc	5	300	Onan Division/Studebaker	$900
115 v 60 cps	5	230	Homelight, Sears	500

B - Gasoline Engines

Maximum Power (hp)	Speed (rpm)	Weight (lb)	Manufacturer	Cost
10	10,000	11	McCulloch	$140
6	3,600	36	Tecumseh Lauson	90
6	3,600	43	Briggs and Stratton	100

C - Generators

Voltage	Maximum Power (Kv-a)	Speed (rpm)	Weight (lb)	Manufacturer	Cost
30 v dc	4.5	7,200	24	Bendix	$800
30 v dc	9	4,000	46	Bendix	---
115 v 440 cps	8	3,800	55	Bendix	900

D - High Frequency Generators

Voltage	Power (Kv-a)	Weight (lb)	Manufacturer	Cost
115 v/1225 cps @ 10,500 rpm	3	50	General Electric	$ 400
1000 v/1600 cps @ 4,800 rpm	4	100	Bogue	1,000
1000 v/1600 cps @ 12,000 rpm	4	25*	Lear Siegler	2,[illegible]00*

*Rough estimate on custom-designed unit.

engine and a light-weight generator since it is undesirable to run the engine at the high shaft speeds required.

Additional equipment that will be required for the piezoelectric system includes a cooling fan, a step-up transformer and, for the directly driven system, voltage control circuitry and an engine speed control system.

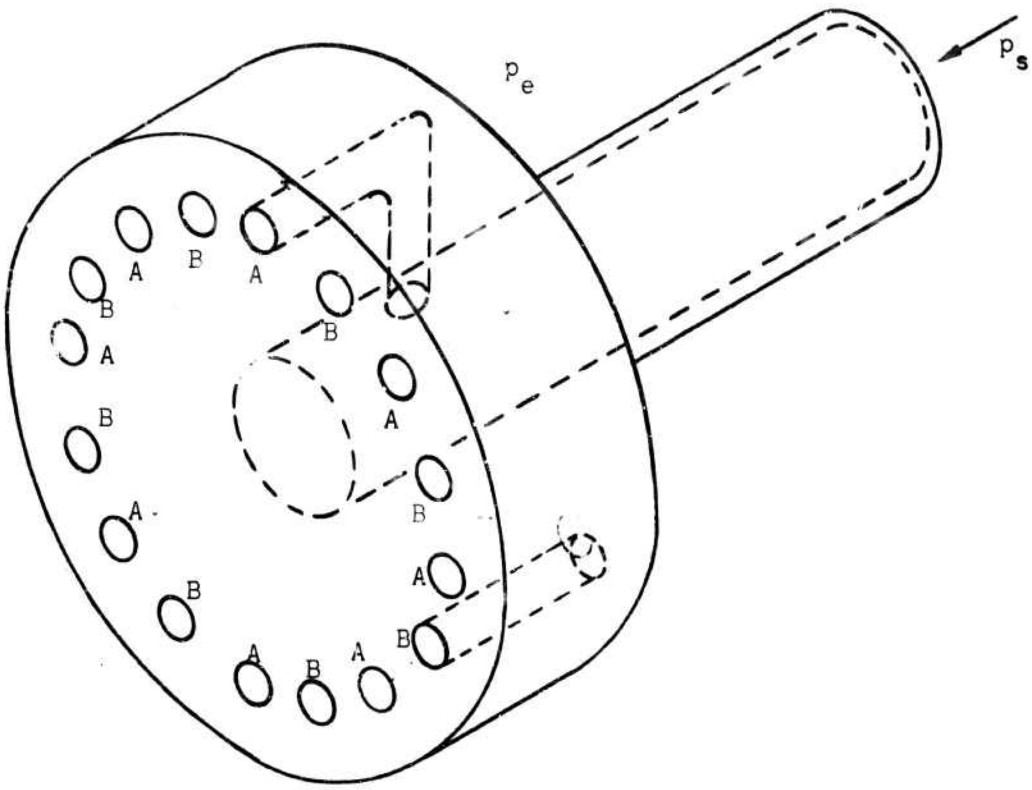
APPENDIX N

LABORATORY MODEL VALVING CONFIGURATION

This appendix describes the geometry of the disk valve and the manifold plate. Pressure unbalance forces have been minimized by using a symmetrical design.

The rotary disk valve is shown schematically in Figure N-1. The fluid to power the piston is introduced through the hollow drive shaft and 8 equally-spaced ports A in the rotating disk. As the disk rotates, ports A register alternately with the ports C and D of the manifold plate shown in Figure N-2. The other 8 ports B in the rotating disk lead the fluid exhausted from the piston through the disk to the oil reservoir. These ports communicate alternately with ports D and C of the manifold plate.

The valve disk with its sixteen holes rotates at a small gap from the fixed manifold plate which has 2 sets of 8 equally-spaced ports C and D. Ports C communicate with the chamber on one side of the piston, ports D with the other. Thus, as the holes in the disk valve sweep by the holes in the manifold plate, the piston chambers are alternately connected to the supply pressure and to the exhaust reservoir.



Ports A are connected to p_s inside the hollow shaft.

Ports B extend through the disk to the exhaust reservoir at pressure p_e.

Figure N-1. Rotating Disk Valve

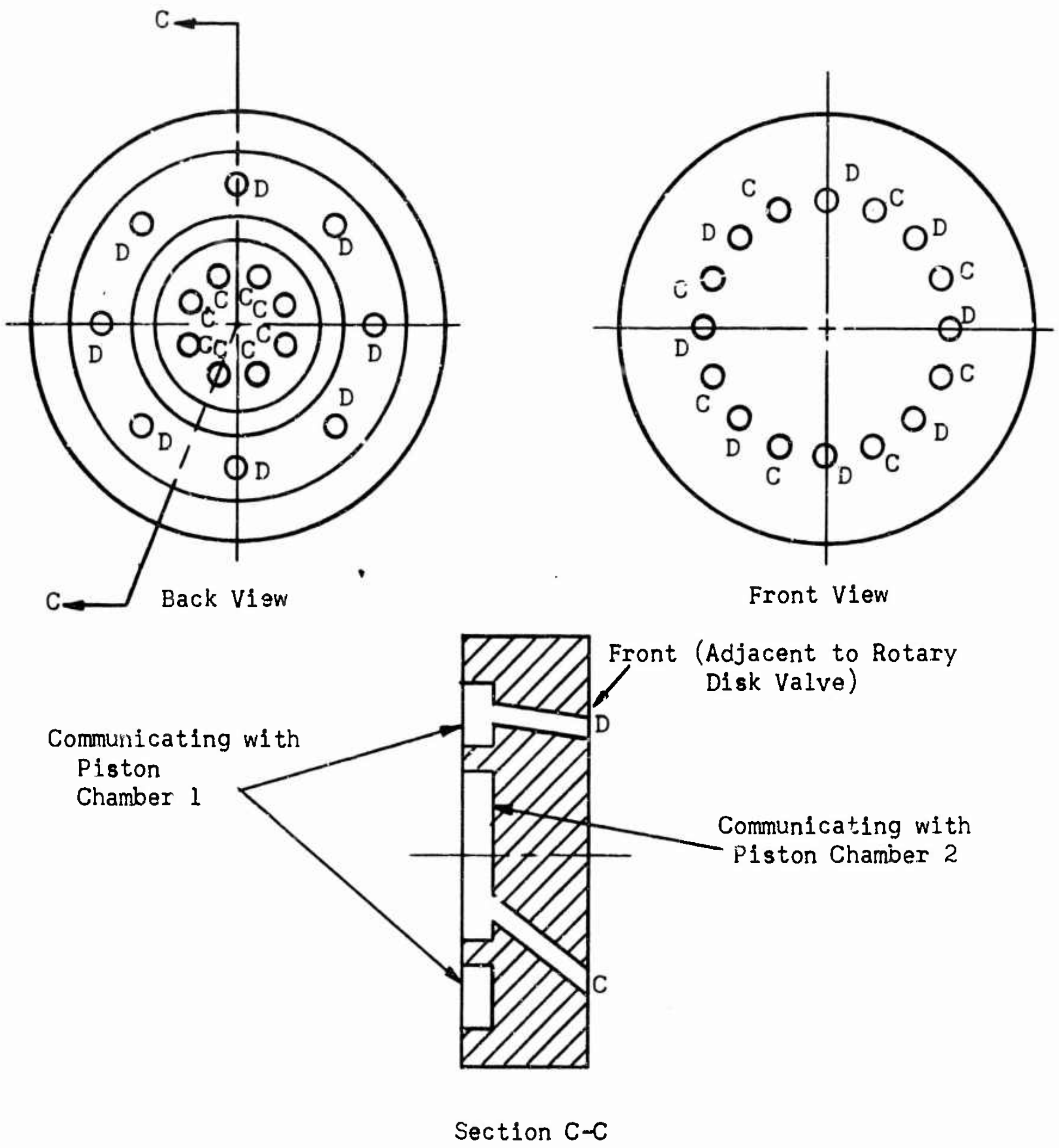


Figure N-2. Manifold Plate for Rotating Valve

UNCLASSIFIED
Security Classification

DOCUMENT CONTROL DATA - R & D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY (Corporate author)
Foster-Miller Associates, Inc.
Waltham, Massachusetts

2a. REPORT SECURITY CLASSIFICATION
UNCLASSIFIED

2b. GROUP

3. REPORT TITLE
An Analytical and Experimental Study to Develop a Nonmechanical System to Induce Resonance in a Rod Drill for Frozen Soil

4. DESCRIPTIVE NOTES (Type of report and inclusive dates)
Final Report: November 4, 1967 to June 4, 1968

5. AUTHOR(S) (First name, middle initial, last name)
Seth R. Goldstein

6. REPORT DATE	7a. TOTAL NO. OF PAGES	7b. NO. OF REFS
June 1969	74	13

8a. CONTRACT OR GRANT NO.
DAAG17-68-C-0028

b. PROJECT NO. 1J662708D503 (formerly 1M624101D503)

c.

d.

9a. ORIGINATOR'S REPORT NUMBER(S)

9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report)
70-18-GP

10. DISTRIBUTION STATEMENT
Each transmittal of this document outside the Department of Defense must have prior approval of the Director, General Equipment & Packaging Laboratory, U. S. Army Natick Laboratories, Natick, Massachusetts 01760.

11. SUPPLEMENTARY NOTES

12. SPONSORING MILITARY ACTIVITY
U. S. Army Natick Laboratories
Natick, Massachusetts 01760

13. ABSTRACT
The objective was to investigate nonmechanical methods for inducing longitudinal resonant vibrations in a rod which could serve as a drill for frozen soil and to select the most promising method for further development.

An analytical model of the mechanics of the vibratory drill was developed to determine the exciter output requirements. A large number of exciter concepts were conceived and evaluated.

The major conclusion is that a hydraulic exciter system is the best overall concept for meeting the exciter output requirements, and its feasibility has been established both analytically and experimentally.

Piezoelectric and magnetostrictive transducers are also feasible but are less desirable than hydraulic systems when weight, durability, safety, cost, and other factors are considered. Unstable bearings, electromagnetic devices, and various other concepts were examined and rejected for reasons of low efficiency and excessive size.

The best application of the hydraulic exciter concept utilizes a rotary valve to control flow to a piston-type actuator that is rigidly attached to the drill rod. A laboratory model of this type of exciter was designed, built, and tested to demonstrate the hydraulic excitation principle. The model system, which used a 6-foot long steel drill rod, exhibited a resonant amplitude peaking of four at a frequency of 1425 cps.

UNCLASSIFIED
Security Classification

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Design	8					
Tests	8					
Drilling machines (tools)	9					
Drills	9					
Frozen soils	4					

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Security Classification